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SCREW PROPELLER DESIGN.

BY CAPTAIN C. W. DYSON, U. S. NAVY, MEMBER.

THE SCREW PROPELLER.

In the study of the action of the screw propeller many theories have been propounded, the principles underlying all of them being comparatively simple. In all of the theories in connection with which mathematical methods are to be used, it is practically necessary to regard the blade as having no thickness.

The three principal English theories of propeller action are those of Professor Rankine, advanced in 1865; of Mr. William Froude, in 1878, and of Professor Greenhill, in 1888.

Rankine's fundamental assumption was that as the propeller advanced with a certain slip, all the water in an elementary ring of radius r was given a certain velocity in a direction perpendicular to the face of the blade at that radius. Then, from the principle of momentum, the thrust from the

elementary ring is proportional to the quantity of water acted upon in one second and to the sternward velocity communicated to it.

Froude considers the element as a small plane moving through the water along a line which makes a small angle with the direction of the plane. He then takes the normal pressure upon the elementary area, which gives propulsive effect to vary as the area, as the square of its speed, and as the sine of the slip angle.

Greenhill attacks the problem from a direction entirely different from that of either Rankine or Froude. He assumes that the propeller is working in a fixed tube with closed end. The result is that the motion transmitted to the water is wholly transverse. The blade is assumed perfectly smooth, so that the pressure produced by the reaction of the water is normal to the blade and has a fore-and-aft component which produces thrust.

In all the above theories the loss by friction is taken as that due to the friction of the propelling plane moving edgewise, or nearly so, through the water.

Naval Constructor D. W. Taylor, U. S. Navy, in his work on "Speed and Power of Ships," criticises these theories as follows:

"On Rankine's theory the water while passing through the screw disc is given sternward velocity. This can occur only if the stream contracts materially while passing through the propeller, or if a material quantity of water from abreast the disc is always flowing into it. Neither motion seems reasonable. Furthermore, on Rankine's theory, the thrust and torque are independent of the blade surface, one assumption of this theory being that the length of the screw and number of its blades are supposed to be adjusted by the rules deduced from practical experience, so that the whole cylinder of water in which the screw revolves shall form a stream flowing aft.

"Practical experience with model propellers shows clearly that the result assumed by Rankine is unattainable. Ran-

kine's theory further ignores variations of pressure which must occur in propeller action.

"Froude's theory goes to the opposite extreme of Rankine's. It assumes that the thrust increases always in direct ratio to the area. Model experiments show conclusively that, while within practicable limits thrust does increase as long as area increases, the increase in thrust is by no means proportional to the area increase, the rate of increase with area diminishing steadily as area increases.

"Greenhill's theory has the same obvious defect as Rankine's, in that it neglects the effect of area of the blade * * * ignores the sternward velocity, deducing thrust entirely from the pressure set up by rotating the water in the disc, but it should be pointed out that his 1888 paper gives some consideration to other possible motions involving axial velocity of slip in the water."

Finally, "As, then, it seems that no theory we have considered can exactly represent the action of propellers, it would be necessary, in case we wished to adhere to formulae, to compare each formula with experimental results and select that one which seemed to agree most closely. Then, using this as a semi-empirical formula, with coefficients and constants deduced from experiments or experience, problems could be satisfactorily dealt with * * * there is obviously no advantage from a practical point of view in attempting to reduce the formulae to forms for use in practice. A serious practical disadvantage is the fact that the formulae use a true slip, based upon true pitch, or a blade of no thickness. The face pitch of a blade with thickness, or its nominal pitch as it may be conveniently called, is very different from the virtual or effective pitch, and this fact causes material complications in using formulae."

The foregoing criticisms on the uses of theoretical formulae in propeller design being well founded, we find ourselves reduced to a choice of any one of the three following methods for the actual work of design:

1. By direct comparison, when all the conditions for a satisfactory vessel of a similar form to the one under consideration are known.

2. By methods based on trials of model propellers in model tanks.

3. By methods based on trials of full-sized propellers in service, over carefully-measured courses.

The first method practically insures a propeller of approximately equal propulsive efficiency with that of the propeller on the compared vessel, but gives no opportunity for improvement in performance.

The second method is open to the decided objection that the conditions under which the model screw is tried in the tank are radically different from those under which the full-sized screw operates. In fact, propellers whose models have shown high tank efficiencies have failed most dismally in service, while other propellers whose models showed poor efficiency, have delivered a high propulsive coefficient. This latter fact has been credited to a high hull efficiency, but a high hull efficiency does not explain why, where models of two or more propellers for the same vessel have been tried, that propeller whose model gave the higher tank efficiency has failed while the propeller with the lower tank efficiency has succeeded.

The third method is open to the objections that results are so easily obscured by inaccuracies in timing runs over the measured course, and in measurements of power and revolutions, in variation of conditions as to smoothness of immersed skin of the ship, surfaces of the propellers, trim and displacement of the vessel, conditions of wind and sea, and the skill of the helmsman.

Believing that when trials are carefully conducted and careful attention is paid towards guarding against the above errors, that such errors as creep in with a full-sized ship and propeller have the minimum of effect, the following method of design has been developed. A preliminary presentation

of it has already been made in this JOURNAL (see August and November, 1910, issues), but the following presentation is much more extended:

METHOD OF DEVELOPMENT OF METHOD.

In Froude's investigation of the action of screw propellers and in Barnaby's development of Froude's results, the model propellers used had a constant form of developed area of blade and a constant ratio of greatest width of development to diameter of the screw.

The form of developed area used was an ellipse, of which the major axis was the radius of the propeller, and this form was used for all values of pitch ratio, $P \div D$, that were experimented with.

The constants obtained by Barnaby are expressed by the following formulae:

$$C_A = \frac{AV^3}{\text{I.H.P.}},$$

$$C_R = \frac{R \times D}{V},$$

in which,

A = Disc area of propeller (including hub) in square feet;
R = Revolutions of propeller per minute;
D = Diameter of propeller in feet;
I.H.P. = Indicated horsepower applied to one screw = 2 × effective or tow-rope horsepower required for the bare hull for the speed S;
V = Speed of water through the screw in knots per hour.

For convenience in computation, in the above formulae the speed of the water through the propeller was replaced by the speed S of the ship in knots per hour, an assumed standard wake of 10 per cent. being applied. A standard thrust deduction of 10 per cent. was also adopted, as representing the addition to tow-rope resistance caused by the action of the propeller on the stream lines of the ship.

Assuming that for propellers identical in all respects except that of developed area, the thrust that can be delivered will vary directly as the developed areas, Barnaby's constants can be represented graphically as a chart, and this has been done, as shown by Plate I.

DESCRIPTION OF THE BARNABY CHART OF DESIGN—PL. I.

At the left of the chart are shown curves of equal values of C_A for varying values of developed area ÷ disc area. On the right, erected on logarithmic abscissae values of C_R, are shown curves of propulsive coefficients and of pitch ratios, $P \div D$.

While for method of design this chart is very rough and ready, as it assumes the E.H.P. for the bare hull as equal to 50 per cent. of the I.H.P. required to propel the vessel, neglects appendage resistance entirely and assumes a standard wake and standard augmentation of resistance of 10 per cent. each, it can readily be used to obtain factors from actual trial results of full-sized vessels and propellers from which a more accurate method of design may be derived.

Also, while for the development of this chart an elliptical form of developed area of blade was used, by analyzing the results of trials we can obtain a series of corrective coefficients which will enable us to use it for any standard form of developed or of projected area that we may adopt.

As all forms of hull have not a wake of 10 per cent., nor is the thrust deduction always 10 per cent., it is necessary to determine from the chart by applying to it the results of actual trials the corrective factor to use in order to allow for the correct wake and thrust deduction factors in future designs. To do this we replace V in the formulae $C_A = \frac{AV^3}{\text{I.H.P.}}$ and $C_R = \frac{R \times D}{V}$ by $V = M \times S$, and then proceed to ascertain from the chart the corresponding value of M, S being the actual speed of the ship in knots per hour.

To illustrate the method pursued we will assume the following problem:

A vessel having the following dimensions, length on load-water line 310 feet, greatest immersed beam 49 feet 2 inches, mean draught 18 feet 9 inches, displacement 4,088 tons, block coefficient .509, makes on trial over the measured mile a speed of 19.51 knots per hour while developing an indicated horsepower of the main engines of 9,581, when making 124.8 revolutions per minute with the propellers. She is fitted with twin screws. Find the value of M from the chart, the diameter of the propeller being 13 feet 6 inches, the pitch ratio 1.39 and the ratio of developed area to disc area .402.

Solution: Assume that M equals unity; that is, that the thrust deduction and the wake have equal values, then

$$A = \frac{\pi}{4} \times \overline{13.5}^2$$

$$V = M \times S = 19.51;$$

$$V^3 = \overline{19.51}^3;$$

I.H.P. = 9,581 for two screws = 4,790.5 for one screw;

$$D = 13.5 \text{ feet};$$

$$R = 124.8;$$

$$\frac{P}{D} = 1.39;$$

$$\frac{H.A.}{D.A.} = .402;$$

$$C_A = \frac{\frac{\pi}{4} \times \overline{13.5^2} \times \overline{19.51}^3}{4{,}790.5} = 221.9;$$

$$C_R = \frac{124.8 \times 13.5}{19.51} = 86.35.$$

Entering the chart (Plate 1) with these values of C_A and C_R, we rise vertically along the ordinate at the left corresponding to a developed area ratio of .402 until it intersects the interpolated curve of $C_A = 221.9$, and from this intersection project over horizontally to the right until we intersect the ordinate

erected from the base at $C_R = 86.35$. Plot this point of intersection. We may then solve the problem, using several other values of M, and plot the points so obtained. Having obtained these several points, a line passed through them will cut at some point the curve of pitch ratio, $P \div D = 1.39$, which is the ratio of the propeller under investigation. This line passing through the series of points will be found to be a straight line, and it will further be found that all such lines are parallel to the line of direction A–B, shown on the Chart.

This parallelism to A–B being true, it is only necessary to obtain the point derived by assuming $M = 1$, and pass a line through it parallel to A–B until it intersects the curve of $P \div D$ which the propeller under examination may have, in this case 1.39.

Project vertically down to the base from this last point of intersection and we obtain the proper value for C_R for this screw, which is in this case $= 83.5$, so that

$$C_R = 83.5 = \frac{R \times D}{M \times S} = \frac{124.8 \times 13.5}{M \times 19.51};$$

$$M = \frac{124.8 \times 13.5}{83.5 \times 19.51} = 1.034.$$

Should the vessel have been given a series of runs over the measured mile at different speeds a series of values of M for a number of points on the derived power curve can be obtained and a curve of M be erected on a proper base, using preferably a base of indicated thrusts in pounds per square inch of disc area of propeller, where

I.T. per square inch of Disc Area =

$$\frac{\text{I.H.P.} \times 33{,}000}{\text{Pitch in feet} \times \text{Revolutions per min.} \times \text{Disc Area in sq. ins.}}$$

$$= \frac{\text{I.H.P.} \times 33{,}000}{P \times R \times \frac{\pi}{4} D^2 \times 144} = \frac{\text{S.H.P.} \times 33{,}000}{.92 \times P \times R \times \frac{\pi}{4} D^2 \times 144} = I.T.$$

On this same base we can erect the corresponding curves of apparent slip, of tip speeds of the propeller and of the propulsive coefficients, where

$$\text{Propulsive coefficient} = P.C. = \frac{\text{Effective horsepower of the hull with all appendages}}{\text{Total indicated horsepower of the engines}}$$

$$= \frac{\text{E.H.P.}}{\text{I.H.P.}} = \frac{\text{E.H.P.} \times .92}{\text{S.H.P.}}$$

On the chart of curves so obtained should be noted the block coefficient of fineness of the vessel on which the propeller is fitted, the projected area ratio of the screw $= P.A. \div D.A.$, and the shape of the projected area.

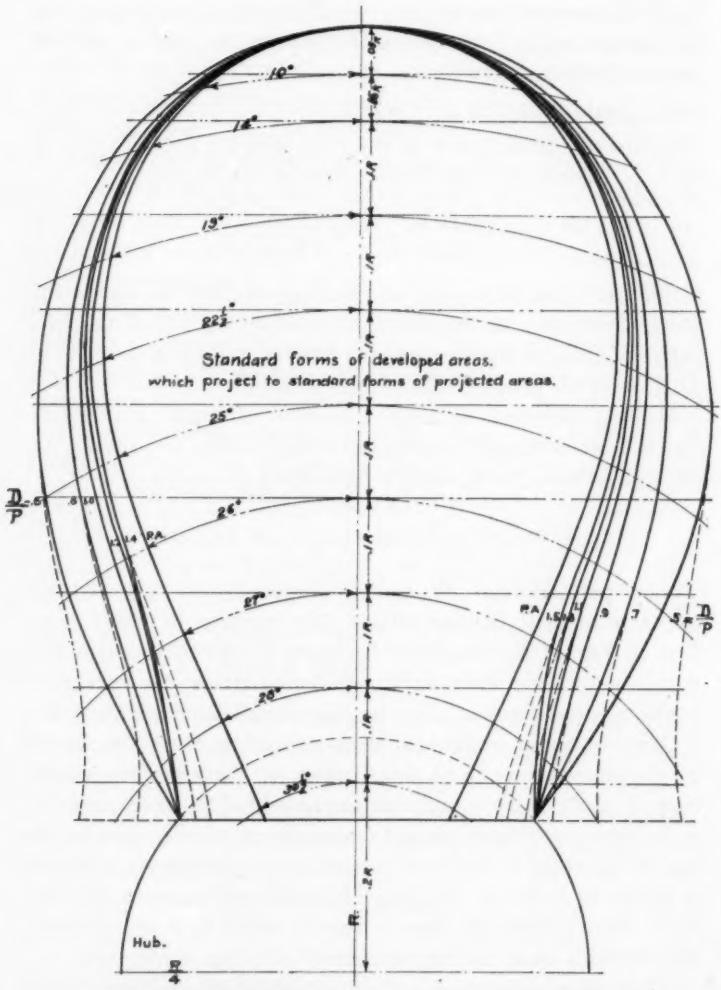
Having collected as large a number of these preliminary charts from as many differently formed vessels as possible, we are in a position to develop a final chart of design.

DEVELOPMENT OF CHART OF DESIGN.

The performances of screw propellers are affected by the form of hull behind which they operate, by their location in regard to this hull, by their pitch, their diameter, their amount and distribution of blade surface, the condition of the blade surface and by the fineness of the blade sections.

Some of these items can be controlled to a certain extent by the designer, such as distribution of blade surface, condition of blade surface and the fineness of the blade sections, and these conditions should be made as nearly constant as may be in order to limit the variables to a minimum. If care is taken to so locate the propeller with reference to the hull as to always insure as free a flow of water to it as possible, this variable may also be considered as being a constant.

There now only remain four variables to consider in the design, namely, form of hull, pitch, diameter and amount of blade surface of the propeller. If a standard form of projected area has been adopted as the basis of design, and all increases or decreases of surface are made proportionally on



Projected area = .32 disc area.

Dotted lines are developments of projected areas as shown.

Full " " actual developments of blades.

For other projected area ratios increase or decrease projected area shown proportionally on circular arcs.

Fig. 1.

the circular arcs of projection, the total developed blade surface will increase or decrease in the same proportion as the projected area, and we can use this latter area in place of the developed area. All of these four variables are mutually dependent, one upon the other, and as any one of them is varied the others must be changed in order to meet these variations.

From an examination of the performances of a large number of propellers the best performances and, in fact, nearly all of the performances that could be rated from very good to excellent, were delivered with propellers having their blade surfaces so distributed as to project on the disc in the general form shown by Figure 1, and for these propellers the greater portion had blade sections similar to or closely approximating the form shown on Figure 2.

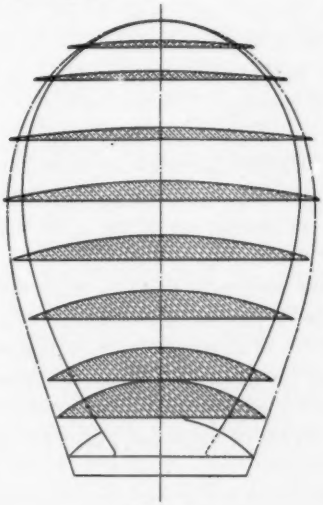


Fig. 2.

The form of projected area shown on Fig. 1 and the section forms shown on Fig. 2 have been adopted as the standard forms. On Plate 22 are given projected-area ratio forms for various ratios, and also a diagram showing the developed-area ratios corresponding to the projected-area ratios.

The very best performances recorded among those collected, both for low and for high values of projected-area ratio, were shown by propellers of this standard form. The ratios giving these performances were $P.A. \div D.A. = .32$ and $P.A. \div D.A. = .54$, and on the results of these screws and on those of other very good to excellent screws of approximately the same form and the same projected-area ratio the Charts of Design have been based.

Taking the data collected from the preliminary charts, the values of M from these charts were laid down graphically for each unit value of indicated thrust in pounds per square inch of disc area, using as abscissae values of block coefficients of the hulls from zero to unity, and as ordinates the values of M. Having laid down these values of M, curves of M passing through the plotted points of M for those propellers approximating closely in projected-area ratio and in projected-area form to the best record screws, that is, those of .32 and .54, were drawn, passing also through the zero value of M, where the block coefficient equalled unity. These curves of M varied but slightly through a wide range of indicated thrusts per square inch of disc area, so mean values of them were laid down as shown on Plate 2 by the curves marked .32 and .54.

With these two curves of M, and with abscissae of M to a large scale and with percentage apparent slip to a small scale as ordinates, curves of apparent slip corresponding to the two curves of M can be drawn. These are then transferred to abscissae of block coefficients and ordinates of apparent slip and a series of curves of apparent slip laid down. These are shown on Plate 3.

CHARTS OF DESIGN.

The final Charts of Design as laid down can be used without correction for the following conditions:

a. Propellers with three blades.

b. Single screws where the dead wood is so cut away as to leave the propeller well clear of the hull.

c. Twin screws where shafts are supported by struts and the blades are given good clearance from the hull.

d. Triple screws embodying the requirements of *b* and *c*, and having shafts separated sufficiently to prevent overlapping of the propeller discs.

e. Any greater number of propellers embodying the requirements of *b*, *c* and *d*.

Corrections to be applied for variations from the above:

a. Where on account of limitations of diameter or for any particular reason it is desirable to use a four-bladed propeller, multiply the E.H.P. of the hull with all appendages, by .865, and use this result as the E.H.P. in the calculations for the propeller. Multiply the projected-area ratio of the propeller thus obtained by $\frac{4}{3}$, and the result will be the necessary projected-area ratio for the four-bladed screw. The propulsive coefficient will probably be very slightly less than that given by the chart for the projected-area ratio of the basic three-bladed propeller obtained by the calculation.

Should a two-bladed propeller be desired, divide the E.H.P. by .75, and take $\frac{2}{3}$ of the projected-area ratio of the basic three-bladed propeller obtained. The propulsive coefficient will probably be slightly greater than that given by the chart for the projected-area ratio of the basic three-bladed propeller.

b. Single screws or screws on the center shaft with multiple screws, when working directly in the wake of and close to the stern post, require a correction for block coefficient in order to select the proper block chart of slips to use in the design. The slip charts as constructed correspond to the Scale of Block Coefficients marked A, on Chart 2. For the conditions under discussion, the block coefficient of the vessel should be taken on the Scale C, and the corresponding reading

on Scale A, vertically over the reading on C will be the correct block chart of apparent slips to use.

Thus, a vessel having a block coefficient of .58 is fitted with a single propeller working in wake of and close to the stern post, the dead wood not being cut away. The reading on Scale A directly over .58 on Scale C is .7, and the .7 block coefficient scale of apparent slips is the one to use.

c. If the vessel were fitted with twin screws and a Lundborg stern, or with screws having the blades working during part of the revolution in very close proximity to the hull, in calculating for the screws so operating, the block-coefficient reading should be first taken on Scale B and transferred to Scale A as before.

The corrections carried by Scale C were obtained as follows: By searching through the records of performances several cases were found where with triple screws having the same pitch, diameter and projected-area ratio for the center screw as for the wing ones, and with the center screw working directly in the wake of and close to the stern post, the shaft horsepower on the three shafts was equally divided, the revolutions of the center shafts in each case being considerably lower than those of the wing shafts. Assuming that with equal powers applied to the shafts, and with equal pitches, diameters and areas, the revolutions of the propellers will vary directly as the velocity of flow of the water through the discs, we obtain at once the correction factor for M to bring us to the condition for the middle screw, and this corrected M being applied to the curve of M for the projected-area ratio of the propellers being considered, gives the new block coefficient to be used for this condition of operation, and from these corrections Scale C was laid down. The condition for screws working behind Lundborg sterns, or with their inner blades working very close to the hull, being approximately a midway condition between A and C, Scale B was so interpolated.

Basic assumptions upon which the charts are constructed.

1. That for any fullness of hull, projected area, pitch and diameter of the propeller are dependent on each other and must vary together. We can, therefore, assume that as the propulsive coefficient for any hull depends on the proper combination of projected area, pitch and diameter for the conditions existing in the wake of the hull, we may take as a base from which to determine the remaining variables, any one of the three; for these charts, the projected area expressed as a fraction of the disc area has been so taken.

2. That with any particular projected-area ratio, the same efficiency of propulsion can be realized, no matter what the fullness of hull, provided the pitch and diameter of the screw are so adjusted as to properly meet the conditions existing in the wake of the hull.

3. That the velocity of the water through the propeller is not only dependent on the form of hull, but also on the propeller itself, and therefore on the projected-area ratio of the propeller. It is on this assumption that Chart 2, curves of M, is built up for various values of projected-area ratios, using the curves of M for $P.A. \div D.A. = .32$ and $= .54$ as the base of construction.

4. That as the tip speed of the screw varies with its pitch and its diameter, and as these vary with the projected-area ratio, the tip speed must vary with this same ratio in a similar manner.

5. That as the apparent slip for any fullness of hull depends on the pitch and the diameter of the propeller, and as these vary with the projected-area ratio, so will the apparent slip vary with it in like manner.

6. That shaft horsepower is equal to 92 per cent. of indicated horsepower. If for any reason there is cause to doubt that the mechanical efficiency of the propelling machinery will be so great, the indicated horsepower obtained from the calculations can be corrected, multiplying it by the factor $\frac{92}{x}$, or the indicated horsepower to be used in the calculation

can be obtained by dividing the designed indicated horsepower of the engines by the same factor, where x is the expected mechanical efficiency of the engines.

DESCRIPTION OF THE CHARTS.

The abscissae of all the Charts of Propulsive Coefficients, of Tip Speeds and of Apparent Slips are "Pounds Indicated Thrust per square inch of Disc Area." Where shaft horsepowers are being dealt with the shaft horsepower is first reduced to indicated horsepower by dividing it by .92, or if we are determining the shaft horsepower necessary for any case the indicated horsepower obtained by the calculation is multiplied by .92.

Plate 4, Chart of Propulsive Coefficients: In the first presentation of this subject the propulsive-coefficient curves presented were the mean curves, and no account was taken of the fluctuations occurring in the coefficients. Neither was any attention paid to the coefficients for thrusts lower than one pound per square inch of disc area. Such low thrusts seldom enter into problems of design, but they do enter into the problem of laying down curves of expected performance for any derived screw.

The curves shown on Plate 4 are obtained from the curves of propulsive coefficients for the chosen basic propellers, and take these fluctuations into account. It may be a mere coincidence, but inspection of the Chart will show that corresponding increases and decreases in the coefficients occur at approximately equal thrusts per square inch of projected area.

The diagonal lines marked 1–1, 2–2, etc., up to 21–21, and shown on all the charts, are lines of equal indicated thrusts per square inch of projected area, from one pound to twenty-one pounds. They are drawn through the points on the curves of propulsive coefficients, Plate 4, curves of tip speeds, Plate 5, and curves of apparent slips, Plates 6 to 20, both inclusive, for projected-area ratios of .32 and .54, corresponding to these indicated thrusts, and the points for the other projected-area ratios are interpolated.

The first high value of propulsive coefficient is seen to be reached at a very low thrust. The second high value, through which the line *H–K* is drawn, is that at which we aim when using low powers, low speeds, a low number of revolutions of propellers, and especially in cases where vessels steam constantly at one speed. After passing the line *H–K* we arrive at a point of critical thrust marked by the line *C–B*, where we find a decrease in the increment of change in propulsive-coefficient values. It is between *H–K* and *C–B* that propellers for moderate powers, speeds and tip speeds will generally fall. After passing *C–B* there is a rapid increase in the rate of decrease of the values of the propulsive coefficients, and when they have reached the line *D–E* we are at practically the upper limit of design for reciprocating-engine propellers, as, if we use any considerably higher values of thrust per square inch of projected area than those corresponding to the line *D–E*, heavy vibrations due to fluctuation of thrust throughout a revolution are apt to occur.

The design of propellers for use with reciprocating engines practically ceases at a projected-area ratio of .43, and it is about at this point that the field of the turbine screws begins, extending to about .54 for moderately high revolutions and high powers, and from .54 to .64 for high revolutions and high powers.

Due to the even torque of turbine-driven shafts, we do not encounter the great fluctuations of thrust during a revolution of the propeller that are met with in reciprocating-engine propellers. This fact permits of a much higher thrust per square inch of projected area being used in the design, so that the ordinary limits of design for such screws may be taken as extending from 19 pounds to 21 pounds indicated thrust per square inch of projected area. The heavy lines *R–S*, *L–M* and *F–G*, shown on all the Charts of Design, correspond to indicated thrusts on the projected area of 19, 20 and 21 pounds per square inch.

Plate 5, Chart of Tip Speeds: This chart is constructed similarly to Chart 4, except the curves of tip speeds for pro-

jected-area ratios of .32 and .54 are used instead of the propulsive coefficient curves. The corresponding lines *H–K*, *C–B*, *D–E*, *R–S*, *L–M* and *F–G* are also shown, together with curves of projected-area ratios corresponding to these lines, and the curves of propulsive coefficients corresponding to *D–E*, *F–G*, *L–M* and *R–S*. This chart is much more convenient for ordinary use than Chart 4, but this latter chart must be used when a series of screws with constant projected-area ratio is being developed and when performance curves are being deduced.

Plates 6 to 20; Charts of Apparent Slips: These charts are constructed in a similar manner to Charts 4 and 5, except that charts are given for each block coefficient of .2, .25, .3, etc., to .9, the curves of apparent slip for projected-area ratios of .32 and .54 being constructed from the apparent slip curves given by Plate 3. The corresponding lines of *H–K*, etc., are also shown. For intermediate block coefficients the apparent slips must be obtained by interpolating from the block charts adjacent to the block coefficient under consideration.

Plate 21, Lines of Thrust and Revolutions: As a guide, and a rough one, to show where to enter the charts for any desired number of revolutions, this chart has been constructed. The abscissae values are expressed by the following:

$$\frac{D^{\frac{1}{3}} S^3}{B.C. \times 10{,}000},$$

where D = Displacement of the vessel in tons, S = Speed in knots per hour and $B.C.$ = the Block Coefficient. The ordinates are pounds per square inch of disc area. Some of the lines have been drawn from actual performances, while the remainder have been interpolated. The method of using this chart will be described in one of the problems in this article.

Tables of Performances: As a further guide in the use of the charts and as a rough check on propellers obtained by calculation, an extensive table of hull data, with the corre-

sponding propellers and performances are appended to this text. In the table of turbine and motor-driven vessels, the shaft horsepowers have been converted into indicated horsepowers, and the shaft thrusts into indicated thrusts.

In cases where there are more than two shafts, the results are given separately for the two wing shafts combined and for the middle shaft alone in the case of a three-shaft vessel, and for the combined inner shafts where four shafts exist. In these cases particular attention is directed to the lower slips and higher thrusts of the middle and inner screws as compared with those quantities for the wing screws.

In the column of "Form of Projected Area," A denotes standard form, B of narrower tip and C of broader tip than standard, while A_B and A_C denote close approximation to standard, but slightly narrower and slightly broader tips, respectively.

Where a propeller is designed by these Charts of Design, blades of B form should never be used. Where we are forced to design with insufficient diameter, blade of A_C or C form should be used, the amount of tip broadening depending entirely upon the decrease in diameter below the calculated diameter which the conditions force upon us.

APPENDAGE RESISTANCE.

The model-tank curves of E.H.P., as usually furnished the designer, are those for the bare hull only. As this is not the true E.H.P. which the propeller is required to deliver, but lacks the E.H.P. necessary to overcome the resistance of the appendages attached to the hull, it becomes necessary to estimate this additional effective power before the problem of the propeller can be approached.

In order to intelligently make this estimate an idea of the effect of each class of appendage must be had, and in order to give such an idea the following table is submitted. In this table reference letters, A, B, C, D and E are used, and their meaning is as follows:

A = Large appendages, not well placed; axes of strut arms fore-and-aft; two shafts, reciprocating engines.

B = Large appendages, carefully placed; axes of strut arms parallel to stream lines; two shafts, reciprocating engines.

C = Moderate-sized struts, appendages carefully placed as in *B*; two shafts, turbine engines.

D = Small-sized struts, appendages carefully placed as in *B* and *C*; three or four shafts, turbine engines.

E = Single-screw ships, appendages carefully placed.

Reference letter.	Per cent. E.H.P. bare hull.	Per cent. E.H.P. struts, shafts, bosses.	Per cent. E.H.P. bilge keels.	Per cent. E.H.P. docking keels.	Per cent. E.H.P. rudder.	Per cent. E.H.P. large scoops.	Total per cent. E.H.P. of bare hull.
A	100	9.4	4.25	5.5	0.85	2.0	122
B	100	8.0	3.05	5.1	0.85	2.0	119
C	100	5.0	3.05	5.1	0.85	2.0	116
D	100	3.0	3.05	5.1	0.85	2.0	114
E	100	0.0	3.05	5.1	0.85	2.0	111

Should any of the above-listed appendages be not fitted to the hull under consideration the total resistance should be decreased by the amount credited to the missing appendages. It is always preferable to slightly overrate the appendage resistance, as, while it slightly detracts from the propulsive efficiency under trial conditions, the propeller obtained by calculation being slightly too large, the propeller will be all the better for service conditions when the vessel is loaded deeper, and higher powers are required for speeds equal to trial speeds than were required at trial draught.

FORMS AND ABBREVIATIONS USED IN COMPUTATIONS.

In putting the foregoing charts to practical use the mathematical formulae necessary are very simple, the terms used having the following definitions:

P = Pitch of the propeller in feet.
D = Diameter of the propeller in feet.
R = Revolutions of the propeller per minute.
$D.A.$ = Disc Area of the propeller, including the hub, in square feet.
$P.A.$ = Projected Area of propeller blades in square feet.
$\frac{P.A.}{D.A.}$ = Projected-Area ratio of the propeller.
S = Speed of vessel in knots per hour.
s = Per cent. apparent slip expressed as a decimal.
I.H.P. = Indicated horsepower applied to one screw.
S.H.P. = Shaft horsepower applied to one screw.
E.H.P. = Effective horsepower required for the hull fitted with all appendages, to be delivered by one propeller.
$T.S.$ = Tip Speed of propeller in feet per minute.
$I.T.$ = Indicated thrust in pounds per square inch of disc area of the propeller.

Total Indicated Thrust exerted by the propeller =

$$\frac{33{,}000 \times \text{I.H.P.}}{P \times R}.$$

Indicated Thrust per square inch of disc area =

$$I.T. = \frac{33{,}000 \times \text{I.H.P.}}{P \times R \times 144 \times \frac{\pi D^2}{4}} = \frac{4 \times 33{,}000}{144 \times \pi} \times \frac{\text{I.H.P.}}{P \times R \times D^2}.$$

$$\therefore\ D^2 = \frac{4 \times 33{,}000}{144 \times \pi} \times \frac{\text{I.H.P.}}{P \times R \times I.T.} = \frac{291.8\ \text{I.H.P.}}{P \times R \times I.T.},$$

and,

$$D = \sqrt{\frac{291.8 \times \text{I.H.P.}}{P \times R \times I.T.}}. \quad . \quad . \quad . \quad . \quad (1)$$

The value of $P \times R$ in equation (1) is obtained from the following equation :

$$P \times R = \frac{101.33 \times S}{1 - s}. \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$R = \frac{T.S.}{\pi D} \quad \ldots\ldots\ldots \quad (3)$$

$$\therefore P \times R = \frac{P \times T.S.}{\pi D},$$

and

$$P = \frac{P \times R \times \pi D}{T.S.}, \quad \ldots\ldots \quad (4)$$

the value of $P \times R$ in equation (4) being obtained from equation (2).

Tables of elements of design should be laid down as shown in the following problems, and the missing quantities calculated and filled into the table, using for the calculations the following logarithmic form :

291.8 .	log 2.46509 .	log (1)	
I.H.P. .	log ——— .	log (2)	
	log ——— .	log (1) + log (2) .	log (3)
$\frac{101.33\,S}{1-s} = P \times R$.	log — .	log (4)	
I.T.	log — .	log (5)	
	log — .	log (4) + log (5) . .	log (6)
D^2			log (3) − log (6)
D	log (7)		$\frac{\log (3) - \log (6)}{2}$
π	log (8)		
	log (7) + log (8) .	log (9)	
		log (4)	
	log (10)	log (9) + log (4)	
T.S.	log (11)		
P	log (10) − log (11)		
R	log (11) − log (9).		

PROBLEMS IN DESIGN.

In order to assist in giving a proper understanding of the methods to be used in handling the Charts of Design a number of problems covering a wide range of conditions of operation are herewith submitted.

Problem 1. Limited diameter, low tip speed.—Required a propeller for a steam launch whose block coefficient is .35. The designed revolutions of the engine are 450 when developing 136 I.H.P. The maximum diameter that can be fitted is 33 inches. There is a single propeller working directly in the wake of and close to the stern post. Having available the model-tank curve of E.H.P., find the speed that can be expected at full power of the engine.

As the conditions under which the propeller operates correspond to the C Scale on Plate 2, the reading on Scale A corresponding to .35 on Scale C is .47, so to obtain the apparent slips for use in the calculations we must interpolate between the Block Coefficient Apparent Slip Charts of .45 and .50. The tip speed corresponding to a diameter of 33 inches and 450 revolutions per minute = 3,888 feet. The lowest tip speeds given on Plate 5 are those for a value of $\frac{P.A.}{D.A.} = .15$, so we will use this curve of tip speeds in the calculation, beginning with the allowable maximum tip speed of 3,888 feet.

$\frac{P.A.}{D.A.}$	$T.S$ from Plate 5.	$I.T.$ from Plate 5.	$P.C.$ from Plate 4.	I.H.P. designed.	E.H.P. = I.H.P. × $P.C.$	S from model curve.	s from slip charts.	101.33 × S.	$P \times R = \frac{101.33 \cdot S}{1-s}$	D from formula.	P from formula.	R from formula.
										feet.	*feet.*	
.15	3,888	1.3	.788	136	107	12.035	.1048	1,219.5	1,362.3	4.734	5.214	261.3
.15	4,000	1.35	.778	136	106	12.03	.108	1,219.0	1,366.6	4.638	4.978	274.5
.15	4,100	1.4	.76	136	103	12.02	.111	1,218.0	1,370.1	4.548	4.775	286.9
.15	4,200	1.475	.745	136	101	12.01	.1158	1,217.0	1,376.4	4.421	4.552	302.4
.15	4,300	1.55	.718	136	98	12.0	.1218	1,216.0	1,384.6	4.30	4.35	318.3
.15	4,500	1.70	.712	136	97	11.995	.133	1,215.5	1,401.9	4.081	3.994	351.0
.15	4,800	2.00	.694	136	94	11.98	.1586	1,214.0	1,442.7	3.708	3.502	412.0
.15	5,000	2.20	.657	136	89	11.97	.1774	1,213.0	1,474.5	3.498	3.316	455.0

Using diameter as abscissae we can erect on this base of diameters curves of pitch and revolutions from which the

characteristics of the primary screw can at once be picked off. We find this propeller to be as follows:

$$\begin{aligned} \text{I.H.P.} &= 136. \\ R &= 450. \\ \frac{P.A.}{D.A.} &= .15. \\ P &= 3.34 \text{ feet.} \\ D &= 3.5 \text{ feet.} \\ s &= .1774. \end{aligned}$$

By the conditions of the problem, however, the diameter cannot exceed 33 inches = 2.75 feet, therefore retaining the same pitch and the same total projected area, but reducing the diameter to 2.75 feet, the final propeller will have

$$\begin{aligned} D &= 33 \text{ inches} = 2.75 \text{ feet.} \\ P &= 3.34 \text{ feet.} \\ \frac{P.A.}{D.A.} &= .15 \times \left(\frac{3.5}{2.75}\right)^2 = .243. \\ s &= .1774 \times \left(\frac{3.5}{2.75}\right)^2 = .2874. \end{aligned}$$

$$\text{Expected speed} = \frac{3.34 \times 450 \times (1 - .2874)}{101.33} = 10.57 \text{ knots.}$$

The projected area will not have the standard A form, but must be of an exaggerated C form, in order to compensate as much as possible for the reduction in diameter.

The large percentage difference between the derived diameter of the primary propeller and the allowed diameter of the actual one, would seem to indicate the desirability of fitting a four-bladed propeller in this case, as with a four-bladed propeller the total power can be absorbed with a smaller diameter. Therefore, in the above table of data, replace the I.H.P. and the E.H.P. by .865 I.H.P. and .865 E.H.P., retaining the same

quantities as originally in the columns of $\frac{P.A.}{D.A.}$, *I.T.*, *T.S.*, *P.C.*, *S* and *s*, and solve for new values of *D*, *P* and *R*.

Having obtained the Pitch and Diameter of this new primary screw, reduce the screw to a diameter of 33 inches, as before. The actual projected-area ratio of the final screw will be the projected-area ratio of the reduced 33-inch diameter propeller $\times \frac{4}{3}$, divided among four blades.

The diameter, pitch, revolutions and projected-area ratio of the four-bladed propeller may be obtained directly from the results obtained for the primary three-bladed screw, as follows:

Taking *P*, D_1 and R_1 from the primary three-bladed-propeller table in the problem, we have—

D_1 (Primary 3-bladed).	4'.734	4'.638	4'.548	4'.421	4'.30	4'.081	3'.708	3'.498
$D_{11} = D_1 \sqrt{.865}$	4.403	4.303	4.23	4.11	3.999	3.795	3.448	3.253
P (Primary 3-bladed) ...	5.214	4.978	4.775	4.552	4.35	3.994	3.502	3.316
$P_{11} = P \times \sqrt{.865}$	4.85	4.63	4.441	4.233	4.046	3.714	3.257	3.084
R_1 (Primary 3-bladed)..	261.3	274.5	286.9	302.4	318.3	351	412	455
$R_{11} = \frac{R_1}{\sqrt{.865}}$	281	295.2	308.5	325.2	342.3	377.4	443	489.1
$\frac{P.A._{11}}{D.A._{11}} = .15 \times \frac{1}{4}$	...	...	...	...	...	...	...	...
$\frac{P.A.}{D.A.} = .15 \times \frac{1}{4} \times \left(\frac{D_{11}}{2.75}\right)^2$	...	...	...	...	...	...	...	...
$P = P_{11}$	...	...	...	...	...	...	...	...
$D = 33'' = 2'.75$	...	...	...	...	...	...	...	...
$R = R_{11} = 450$	...	...	...	...	...	...	...	...

It will be found, however, that the four-bladed propeller will in such cases usually have excessive area, and it will be preferable to use the three-bladed screw. In special cases, such as tramp steamers and colliers, where there is great variation in loading, resulting in the propellers operating at times when only partially submerged, four-bladed propellers are preferable, in order to prevent too great a variation of resistance of the screw during its revolution.

Problem 2. High speed of vessel, very high revolutions, small diameter of propeller.—Gasoline launch with a block coefficient of .23. Brake horsepower of the engine at 1,000 revolutions per minute = 75. Single screw working well clear of the hull, the dead wood being cut away. Maximum

allowed diameter = 25 inches. The conditions under which the propeller operates are those corresponding to Scale A on Plate 2, so we interpolate between Block Coefficient Charts of Apparent slip, of .20 and .25 for the apparent slips to use in the calculations. The E.H.P. curve of the hull with all appendages is available for use. The tip speed corresponding to 25 inches diameter and 1,000 revolutions = 6,545 feet per minute. Design on line *D–E*.

T.S., Plate 3, Line *D–E*.	*I.T.*, Plate 5.	$\frac{P.A.}{D.A.}$, Pl. 5.	*P.C.*, Pl. 5.	I.H.P. = $\frac{75}{.92}$	E.H.P. = I.H.P. × *P.C.*	*S* from mod. curves.	*s* from slip charts.	101.33 × *S*.	$\frac{P \times R = 101.33 \times S.}{1 - s}$	*D* from formula.	*P* from formula.	*R* from formula.
										feet.	*feet.*	
6,545	3.65	.255	.66	80	52.8	18.9	.193	1,915.14	2,373.2	1.642	1.87	1,269
6,200	3.25	.23	.667	80	53.4	18.92	.191	1,917.16	2,369.8	1.741	2.091	1,134
5,900	3.00	.215	.672	80	53.8	18.94	.190	1,919.19	2,369.4	1.812	2.286	1,036
5,700	2.80	.20	.675	80	54.0	18.95	.189	1,920.2	2,367.7	1.876	2.449	967
5,500	2.60	.19	.679	80	54.3	18.96	.188	1,921.22	2,366.0	1.95	2.635	898
5,300	2.40	.175	.683	80	54.6	18.97	.187	1,922.23	2,364.4	2.028	2.843	832

Laying down curves of *P*, *R* and $\frac{P.A.}{D.A.}$, with values of *D* as abscissae, we find the propeller for 1,000 revolutions to be as follows:

$$D = 1.85 \text{ feet};$$
$$P = 2.375 \text{ feet};$$
$$\frac{P.A.}{D.A.} = .2085.$$

As the diameter of this screw is less than the allowed diameter, no correction for form of blade is required, and the standard form A is used.

Problem 3. Problem in reduced diameter.—Maximum possible efficiency required at cruising speed. Twin-screw collier, constant steaming speed of 13 knots. Ship fitted with Lundborg stern. Revolutions of propellers for this speed = 100. Maximum allowed diameter of screw = 15 feet. Maximum tip speed of screw = 4,712 feet. Block coefficient of

hull on Scale B, Plate 2, = .75; equivalent block coefficient on Scale A = .81. E.H.P. curve with all appendages available. Find propeller and I.H.P. required.

Line of Design.	*H-K*	*C-B*	*D-E*	*H-K*	*C-B*	*D-E*	*H-K*	*C-B*	*D-E*
$T.S._1$, from Plate 5	4,500.	5,200.	5,600.	4,900.	5,850.	6,300.	5,300.	6,600.	7,100.
$T.S.$, allowed	4,712.	4,712.	4,712.	4,712.	4,712.	4,712.	4,712.	4,712.	4,712.
$\frac{P.A.}{D.A.}$ for $T.S._1$, Pl. 5	.2	.2	.2	.25	.25	.25	.3	.3	.3
$I.T._1$ for $T.S._1$, Pl. 5	1.5	2.2	2.8	1.8	2.85	3.55	2.05	3.65	4.4
$T.S. \div T.S._1 = A$	1.047	.906	.841	.962	.805	.748	.889	.714	.664
s_1 from slip charts for $B.C. = .81$	.028	.0566	.078	.0312	.0598	.0794	.036	.0638	.0808
$s = s_1 \div A^2$	.0268	.069	.1106	.0337	.0923	.1419	.0456	.1575	.1833
S designed, knots	13	13	13	13	13	13	13	13	13
$101.33 \times S$	1,317.29	1,317.29	1,317.29	1,317.29	1,317.29	1,317.29	1,317.29	1,317.29	1,317.29
$P \times R = \frac{101.33 \times S}{1-s}$	1,353.6	1,414.1	1,481.1	1,363.2	1,451.2	1,535.1	1,380.2	1,363.6	1,612.9
$S_1 = \frac{P \times R \times (1-s_1)}{101.33}$	12.98	13.17	13.48	13.03	13.47	13.95	13.13	14.45	14.63
E.H.P. per screw for S_1 from E.H.P. curve	1,412.	1,500.	1,651.	1,450.	1,650.	1,950.	1,475.	2,300.	2,400
$F.C.$ for $I.T._1$, Pl. 4	.773	.704	.676	.747	.688	.661	.681	.670	.646
I.H.P. = E.H.P. $\div$ $P.C.$ (1 screw)	1,827.	2,130.	2,442.	1,941.	2,398.	2,950.	2,166.	3,433.	3,715.
D_1 in feet (calc.)	16.2	14.18	13.11	15.19	13.31	12.57	14.95	13.25	12.76
$P_1 = P$ in feet (calc)	15.32	12.03	10.89	13.28	10.37	9.621	12.23	9.861	8.82
$R_1 = R$ (calculated)	88.4	116.7	136.	102.7	139.9	159.9	112.9	158.6	182.9
E.H.P. for 13 knots	1,415.	1,415.	1,415.	1,415.	1,415.	1,415.	1,415.	1,415.	1,415.
$P.C.$ of final screw, 13 knots = $\frac{\text{E.H.P.}}{\text{I.H.P.}}$									
13 knots	.775	.662	.577	.726	.588	.478	.651	.4107	.378

Laying down curves of P, R, $\frac{P.A.}{D.A.}$, and of $P.C.$ on diameter as abscissae, we find the following two propellers, one having the allowed diameter and the other giving the desired revolutions:

	First propeller.	Second propeller.
$\frac{P.A.}{D.A.}$	.20	.2375
D	15 feet.	15.36 feet.
P	13.4 feet.	13.625 feet.
R	104.5	100
$P.C.$	.713	.74.
I.H.P. on 1 screw,	1,985	1,912
I.H.P. on 2 screws,	3,970	3,824
Speed,	13	13

Modified diameter of 2d screw, . . . 15 feet.

Modified $\frac{P.A.}{D.A.}$ of 2d screw $= .2375 \times \left(\frac{15.36}{15}\right)^2 = .25.$

Apparent slip of 2d screw, ...0332.

Modified slip of 2d screw, $= .0332 \times \left(\frac{15.36}{15}\right)^2 = .0348.$

Modified speed of ship, ... 12.98.
E.H.P. for modified speed, . . . 1,412.
P.C. of modified 2d screw, ...738.

Use modified 2d propeller with A_C form of projected area on account of the reduced diameter as compared with that of the primary screw.

Problem 4. Vessel of moderate power and speed, moderate number of revolutions.—Find the power and the propeller necessary to propel a battleship having a block coefficient of .65, at a speed of 17 knots per hour. The revolutions of the propellers for this speed to be 120. Maximum allowed diameter 17 feet. Twin screws working well clear of the hull. Maximum tip speed corresponding to 120 revolutions and 17 feet diameter = 6,409. E.H.P., with all appendages, at 17 knots = 8,000. Design on line *D*–*E*, using Scale A of Plate 2.

T.S., Pl. 5.	*I.T.*, Pl. 5.	$\frac{P.A.}{D.A.}$ Pl. 5.	*P.C.*, Pl. 5.	E.H.P. on 1 screw.	I.H.P. = E.H.P. ÷ *P.C.*	*S.*	*s* from slip chart for .65.	101.33 *S.*	$\frac{P \times R = 101.33\ S}{1 - s}$	*D*, calc.	*P*, calc.	*R*, calc.
										feet.	*feet.*	
6,409	3.5	.247	.663	4,000	6,034	17	.1285	1,722.6	1,976.6	15.95	15.46	127.8
6,200	3.3	.235	.666	4,000	6,006	17	.128	1,722.6	1,975.5	16.32	16.34	120.9
6,000	3.05	.218	.670	4,000	5,970	17	.127	1,722.6	1,973.2	17.01	17.58	112.3
5,800	2.9	.210	.673	4,000	5,944	17	.1265	1,722.6	1,972.1	17.41	18.61	106.
5,600	2.65	.190	.678	4,000	5,900	17	.126	1,722.6	1,970.9	18.16	20.07	98.18

Plotting the curves of *P*, *R*, $\frac{P.A.}{D.A.}$, and of *P.C.*'s, as before, we obtain the necessary propeller and power as follows:

Revolutions, 120;
I.H.P. (two screws), 12,005; Speed, 17 knots;
$D = 16.39$ feet; Slip, .1278;
$P = 16.45$ feet;
$\frac{P.A.}{D.A.} = .2331$;
Form of Projected Area, *A*.

Problem 5. Battleship of high power and speed, moderate revolutions, twin screws.—Block coefficient = .615. Speed = 21 knots. Revolutions of propellers = 125 per minute. E.H.P. of bare hull for the required speed = 15,600. Estimated appendage resistance = 16 per cent. of bare hull resistance. ∴ Total estimated E.H.P. = 15,600 × 1.16 = 18,100 = 9,050 on one screw. E.H.P. curve on bare hull is available for use in plotting expected performance of ship. Calculate necessary propeller and I.H.P. for 21 knots, also estimated curves of speed-revolutions and speed — I.H.P. Maximum allowable diameter = 19 feet. Tip speed corresponding = 7,461. Conditions as per Scale A, Plate 2. Design on line *D–E*.

T.S., Plate 5.	*I.T.*, Plate 5.	$\frac{P.A.}{D.A.}$ Pl. 5.	*P.C.*, Pl. 5.	E.H.P.	$\text{I.H.P.} = \frac{\text{E.H.P.}}{P.C.}$	*S.*	*s* from slip charts.	101.33 *S.*	$\frac{P \times R = 101.33\,S.}{1 - s}$	*D*, calc.	*P*, calc.	*R*, calc.
										feet.	*feet.*	
7,461	4.725	.32	.64	9,050	14,141	21	.142	2,127.93	2,480.1	18.76	19.6	126.6
7,300	4.5	.307	.644	9,050	14,053	21	.141	2,127.93	2,477.2	19.18	20.45	121.2
7,000	4.2	.291	.648	9,050	13,966	21	.1385	2,127.93	2,470.0	19.82	21.97	112.4
6,700	3.85	.270	.655	9,050	13,817	21	.138	2,127.93	2,468.6	20.6	23.84	103.6
6,400	3.50	.250	.662	9,050	13,671	21	.136	2,127.93	2,462.9	21.51	26.01	94.7

Plotting the derived data, as before, we take off from the curves the propeller and power required, which are—

Revolutions 125, I.H.P. (two screws) 28,252.
Diameter, 18.85 feet; Pitch, 19.8 feet.
$\frac{P.A.}{D.A.} = .317$, Apparent slip = 14.1+ per cent.
Blades of Standard Projection *A*.

To estimate the performance of this propeller we have at command the E.H.P. curve on the bare hull of this vessel, and this curve must be first corrected throughout for the additional appendage resistance. Having prepared this corrected curve, we proceed as shown in the following table:

$T.S.$ assumed for $\frac{P.A.}{D.A.} = .317$.	$I.T.$ for these values of $T.S.$ from Plate 5.	Revs. $= \frac{T.S.}{\pi D}$.	Total Ind. Thrust $= \frac{33{,}000 \times \text{I.H.P.}}{P \times R.} = \frac{\frac{1}{2} \pi D^2}{144} \times I.T.$	I.H.P. in 1 screw $= \frac{\text{Total } I.T.}{33{,}000}$.	$P.C.$ for $I.T.$'s in Col. 2, taken from $P.C.$ curve of Plate 4.	Total E.H.P. $=$ I.H.P. $\times P.C. \times 2$.	Speed for total E.H.P. from corrected speed — E.H.P. curve.	Slips $= \frac{[(P \times R) - (101.33 \times S)] \div (P \times R)}{}$.
3,400	1.	57.4		1,382	.71	1,962	10.2	
4,500	1.5	76.		2,719	.684	3,720	12.92	
5,250	2.	88.6		4,265	.708	6,040	15.10	
5,825	2.5	98.4		5,921	.712	8,432	16.67	
6,300	3.	106.4		7,683	.692	10,634	17.98	
6,700	3.5	113.1		9,528	.671	12,786	19.13	
7,000	4.	118.2		11,380	.664	15,114	20.06	
7,300	4.5	123.3		13,360	.648	17,308	20.75	
...	...	125.		14,126	.6406	18,100	21.00	

Having the necessary data calculated, the required curves can be laid down, using either revolutions or speed as abscissae, as may be desired. Generally abscissae of revolutions is preferable.

Problem 6. A destroyer with reciprocating engines.—I.H.P. of two engines = 8,400. Twin screws. Revolutions at full power = 335 per minute. The block coefficient of the hull = .41. The maximum allowable diameter = 8 feet, the corresponding tip speed at 335 revolutions being 8,420 feet. The propellers are carried well clear of the hull, making conditions fit Scale A, Plate 2. The E.H.P. curve of hull, with all appendages, is available for use. Design on line *D–E*.

$T.S.$	$I.T.$, Pl. 5.	$\frac{P.A.}{D.A.}$, Pl. 5.	$P.C.$, Pl. 5.	I.H.P. 1 screw.	E.H.P. $=$ I.H.P. $\times$ .92.	$S.$	s from slip chart.	101.33 $S.$	$P \times R = \frac{101.33 \times S}{1 - s}$	D, calc.	P, calc.	R, calculated.
										feet.	*feet.*	
8,420	5.92	.387	.621	4,200	2,608	31.3	.181	3,171.6	3,872.5	7.312	10.56	366.5
8,260	5.75	.378	.623	4,200	2,617	31.5	.181	3,191.9	3,897.3	7.395	10.96	355.5
8,050	5.5	.363	.627	4,200	2,633	31.7	.181	3,202.2	3,909.8	7.55	11.52	339.4
7,800	5.2	.346	.632	4,200	2,654	31.9	.181	3,232.4	3,946.8	7.728	12.28	321.3

Plotting these results as before, we can at once take from the obtained curves the required characteristics of the propellers necessary to efficiently absorb the power of the engines.

Problem 7. Destroyer, three shafts, turbine engines.—Revolutions of turbines for guaranteed water rate per shaft horsepower = 700. Speed of vessel = 30 knots. Available power provided for this speed = 15,500 equally divided among the three shafts. Block coefficient of vessel = .412. Wing screws work well clear of hull in Scale A, Plate 2, conditions. Center screw works directly in wake of and close to the stern post. Resultant block coefficient for apparent slips to use for this condition, referring .412 on Scale C back to Scale A, is .55. E.H.P. of hull, with all appendages, for 30 knots is 8,690.

$$\text{Total I.H.P.} = \frac{15{,}500}{.92} = 16{,}848 \text{ on three shafts.}$$

$$= 5{,}616 \text{ on each shaft.}$$

$$\text{Necessary } P.C. \text{ for 30 knots} = \frac{8{,}690}{16{,}848} = .516.$$

W = Wing screws. C = Center screw.

Necessary $P.C.$	$\frac{P.A.}{D.A.}$, Pl. 4.	$I.T.$, Pl. 4.	$T.S.$, Pl. 5.	E.H.P. one screw for 30 knots	E.H.P. $\div$ .92 = I.H.P.	S = speed.	s from .412 and from .55.	101.33 $\times$ S.	$P \times R = \frac{101.33 S}{1-s}$	D by calculation.	P by calculation.	R by calculation.
.516	.60	11.875	12,720	2,897	5,616	30	.1937W .1755C	3,039.9	3,770.2W 3,685.7C	6.05 6.119	5.633 5.569	669.2W 661.9C
.516	.61	12.07	12,900	2,897	5,616	30	.1905W .173 C	3,039.9	3,755.3W 3,675.8C	6.013 6.077	5.499 5.44	682.9W 675.7C
.516	.62	12.275	13,050	2,897	5,616	30	.1882W .171 C	3,039.9	3,744.6W 3,666.9C	5.971 6.034	5.382 5.326	695.7W 688.4C
.516	.63	12.475	13,200	2,897	5,616	30	.1857W .1682C	3,039.9	3,733.2W 3,642.6C	5.932 6.005	5.271 5.206	708.3W 699.6C

Plotting separate sets of curves for the wing and the center screws, we find from them the propellers, as follows:

	Wing.	Center.
Diameter,	71½ inches,	72 inches.
Pitch,	64 inches,	62½ inches.
$P.A. \div D.A.$,	.627,	.63.
Revolutions,	700	700
S.H.P.,	15,500	
Speed,	30 knots.	

Problem 8. Battleship with turbine engines. Three shafts.—Power equally divided. Dead wood cut away, so that there is practically as free a flow of water to the middle screw as to the wing ones. Designed shaft horsepower of main engines = 45,300. Designed revolutions for this power = 277. Block coefficient of hull = .64. Designed speed = 22.5 knots. E.H.P., with all appendages for this speed = 25,000. E.H.P., with all appendages for 15 knots = 5,250. Deduce the propellers for the designed speed, revolutions and S.H.P., and find the S.H.P. necessary for 15 knots.

Conditions as per Scale A, Plate 2.

$$\text{S.H.P.} = 45{,}300 \therefore \text{I.H.P.} = \frac{45{,}300}{.92} = 49{,}240 \text{ on 3 shafts.}$$
$$= 16{,}414 \text{ on 1 shaft.}$$

$$P.C. \text{ necessary to obtain for 22.5 knots} = \frac{25{,}000}{49{,}240} = .508.$$

Necessary $P.C.$	$\frac{P.A.}{D.A.}$, Pl. 4.	$I.T.$, Pl 4.	$T.S.$, Pl. 5.	E.H.P. on 1 screw.	I.H.P. on 1 screw.	$S.$	s from chart.	$101.33 \times S.$	$\frac{P \times R = 101.33 \ S.}{1 - s}$	D by calculation.	P by calculation.	R by calculation.
.508	.50	9.95	11,000	8,334	16,414	22.5	.182	2,279.93	2,787.2	13.14	10.36	269.
.508	.52	10.35	11,400	8,334	16,414	22.5	.180	2,279,93	2,780.4	13.05	10.00	278.
.508	.54	10.74	11,748	8,334	16,414	22.5	.174	2,279.93	2,760.2	12.71	9.382	294.2
.508	.55	10.97	11,950	8,334	16,414	22.5	.173	2,279,93	2,756.9	12.58	9.121	302.3

And we find the propeller required to have the following particulars:

$$D = 13.075, \ P.A. \div D.A. = .519.$$
$$P = 10.10, \text{ Revs.} = 277.$$

To estimate the S.H.P. for 15 knots, proceed as in Problem 5, and as shown in the following table:

$I.T.$ from Plate 5.	$T.S.$ from Plate 5, curve .519.	Revs. $= \frac{T.S.}{\pi D}$	Total thrust $= \frac{I.T. \times 144 \times}{\frac{\pi D^2}{4} \times P \times R}$	I.H.P. = Total Ind. Thrust ÷ 33,000.	$P.C.$ for $I.T.$ in Col. 1, taken from .519 curve, Plate 4.	E.H.P., one screw = I.H.P. $\times P.C.$	Total E.H.P. = 3 × E.H.P.	S.H.P. = I.H.P. × .92.	Total S.H.P. = 3 × S.H.P.
2.75	6,750	164.5	...	2,650	.613	1,628	4,884	2,438	7,314
3.00	7,000	170.3	...	2,993	.621	1,858	5,574	2,753	8,259
3.25	7,250	176.7	...	3,373	.624	2,105	6,315	3,103	9,309
3.50	7,400	180.3	...	3,696	.624	2,306	6,918	3,400	10,200

Plotting curves of S.H.P. and of revolutions on E.H.P. as abscissae, we find the S.H.P. and revolutions corresponding to 5,250 E.H.P., and 15 knots, to be 7,800 and 167.5, respectively.

Problem 9.—In order to show the method of using Plate 21, the following problem is given: Battleship of 20,000 tons displacement. Designed speed, 21 knots. Block coefficient, .60. Four shafts, with all propellers well clear of the hull and of each other. This is a Scale A, Plate 2, condition. Designed S.H.P. for this speed = 28,000, with 330 revolutions. E.H.P. of hull, with appendages = 15,400.

$$\text{Total I.H.P.} = \frac{28{,}000}{.92}, \text{ I.H.P. on 1 shaft} = 7{,}609.$$

$$P.C. \text{ necessary for 21 knots} = \frac{15{,}400 \times .92}{28{,}000} = .506.$$

$$\frac{\overline{\text{Displacement}^{\frac{2}{3}}} \times \text{Speed}^3}{\text{Block coefficient} \times 10{,}000} = 11{,}370.$$

Entering Plate 21, with an abscissa value of 11,370, we see by the scale of ordinates that for approximately 330 revolutions at this value of abscissa an approximate *I.T.* of 9.75 pounds per square inch of disc area must be used. Entering Plates 4 and 5, with this value of *I.T.*, and taking the *T.S.* and the *P.A.* ÷ *D.A.* corresponding to a *P.C.* of .506 for this value of *I.T.*, we calculate the top screw in the following table. The resultant screw, having greater revolutions than are desired, indicates that the *I.T.* value used was too high, so we proceed to calculate a series of propellers with lower values of *I.T.*, but all having the same .506 value of *P.C.*, until we have reached or passed that one whose revolutions are 330, when we construct curves as before and pick off the necessary propeller.

P.C.	*I.T.*	*P.A.* / *D.A.*	*T.S.*	I.H.P.	E.H.P.	*S.*	*s.*	101.33 *S.*	*P* × *R.*	*D.*	*P.*	*R.*
.506	9.75	.49	10,900	7,609	3,850	21	.195	2,127.93	2,643.4	9.282	7.071	373.8
.506	9.5	.476	10,750	7,609	3,850	21	.197	2,127.93	2,650.0	9.391	7.273	364.4
.506	9.25	.464	10,500	7,609	3,850	21	.20	2,127.93	2,659.9	9.50	7.56	351.8
.506	9.00	.453	10,380	7,609	3,850	21	.201	2,127.93	2,663.2	9.714	7.83	340.1
.506	8.75	.44	10,100	7,609	3,850	21	.206	2,127.93	2,680.0	9.73	8.112	330.4

The bottom screw in the table is the propeller to adopt.

There are many problems which may arise, but the foregoing solutions are considered sufficient to give a fair working knowledge of the charts.

NOTE.—Since preparing the foregoing article the U. S. S. *Utah* has run her preliminary acceptance trials with the following results:

Speed, knots.	Revolutions of shafts.		Slip.		I.H.P.		Propulsive coefficient.	
	Outboard.	*Inboard.*	*Outboard.*	*Inboard.*	*Outboard.*	*Inboard.*	*Bare.*	*App.*
21.042	327.308	319.791	23.454	21.653	14,736	14,640	44.99	55.84
12.018	156.664	188.437	8.57	23.98	1,198	3,083	45.0	51.98

The details of the propellers are given in the data tables. Those of the hull are as follows:

Length on L.W.L., feet, . . .	510
Beam, feet and inches, . . .	88–02$\frac{5}{8}$
Draught (mean), feet and inches, .	27–10$\frac{3}{16}$
Displacement, tons, . . .	21,284
Area immersed M.S., square feet, .	2,458
Block coefficient of fineness, . .	.5837

The results of these trials point out very strongly an important error in propeller design for naval vessels where high speed is required at very infrequent intervals, and that is that in the desire to make a highly economical showing at high speeds the economy at low speeds is deliberately sacrificed.

This sacrifice would be justified in the case of a trans-Atlantic liner, but here the case is entirely different. By sacrificing some efficiency of propulsion at the high speed, and such sacrifice would mean but a very small decrease in maximum speed, not sufficient to put a vessel in a fight nor to take her out of it, a much higher propulsive efficiency at the low speeds would be realized and the cost of operating at these speeds would be much reduced.

Form of P.A. A=Standard; B=Narrower at

Vessels.	Characteristics of hull.						Performance.		
	Displacement.	Length on L.W.L.	Greatest immersed beam	Mean draught.	Area midship section.	Block coeff.	Speed.	Revolutions per minute.	I.H.P. on all screws.
	tons.	*ft. ins.*	*ft. ins.*	*ft. ins.*	*sq. ft.*		*knots.*		
Indiana	10,225	348-0	69-3	23-10½	1,545	.622	15.55	131	9,498
Massachusetts	10,555	348-0	69-3	24-10¼	1,545	.622	16.7	136.5	11,200
Oregon	10,250	348-0	69-3	24-0	1,545	.620	16.8	128.25	10,890
Iowa	11,363	360-0	72-2⅝	24-0½	1,635	.630	17.087	109.56	11,835
Maine (1)	5,500	318-0	57-0	18.6	906	.574	17.45	124.13	9,172
Texas	6,320	301-4	64-0	22-7¼	1,305	.555	17.8	125.25	8,423
Kentucky	11,538	368-1	72-2¼	23-5¼	1,618	.643	16.897	114.	12,082
Kearsarge	11,738	368-1	72-2¼	24-1¼	1,618	.644	16.82	114.05	11,674
Illinois	11,738	368-1	72-2¼	24-1¼	1,618	.644	17.45	117.9	12,647
Alabama	11,734	368-1	72-2¼	23-11¾	1,610	.643	17.0	114.65	11,073
Wisconsin	11,500	368-1	72-2¼	23-11	1,619.6	.642	17.174	119.2	12,322
Maine (2)	12,804	388-0	72-2¼	24-4¼	1,606	.654	18.0?	127.91	15,222
Missouri	12,240	388-0	72-2¼	23-6	1,606	.654	18.145	116.8	15,650
Ohio	12,480	388-0	72-2¼	23-5⅝	1,606	.654	17.82	123.8	15,950
Georgia	14,907	435-0	76-2¼	23-8	1,730	656	19.0	118.4	20,800
Rhode Island	14,860	435-0	76-2¼	23-7⅞	1,730	.655	19.0	125.4	21,260
New Jersey	14,870	435-0	76-2¼	23-8¼	1,730	.656	19.0	124.4	21,260
Nebraska	14,900	435-0	76-2¼	23-10¼	1,730	.659	19.0	119.9	20,700
Virginia	14,978	435-0	76-2¼	23-9	1,730	.658	19.0	125.2	22,450
Connecticut	16,377	450-0	76-10	24-11⅜	1,809	.660	18.5	125.6	18,250
Louisiana	16,033	450-0	76-10	24-7	1,809	.660	18.0	119.0	15,700
Minnesota	16,016	450-0	76-10	24-6⅜	1,809	.660	18.5	119.3	17,300
New Hampshire	16,023	450-0	76-10	24-6½	1,809	.660	18.5	122.0	18,000
Vermont	15,986	450-0	76-10	24-5⅞	1,809	.660	18.5	118.4	18,500
Kansas	16,075	450-0	76-10	24-6⅞	1,809	.660	18.0	120.6	19,000
South Carolina	16,024	450-0	80-2⅜	24-6	1,890	.637	19.3	126.5	19,975
Michigan	16,065	450-0	80-2⅜	25-0½	1,890	.633	19.5	125.0	18,300
Delaware	20,099	510-2½	85-2½	27-0½	2,244	.600	21.42	126.75	26,500
Mississippi, 1st screw	13,040	375-0	77-0	24-8½	...	.640	17.0	120.6	12,850
Mississippi, 2d screw	13,000	375-0	77-0	24-7	...	.640	17.0	124.4	12,550
Saratoga	8,480	380-6½	64-10	22-6½	1,350	.509	21.0	134.82	16,947
Brooklyn	8,150	400-6	64-8⅜	21-10½	1,225	.520	21.91	130.5	18,248
Pennsylvania	13,817	502-0	69-6	24-1.18	1,596.6	.558	22.0	124.6	24,400
Colorado	13,670	502-0	69-6	23-11	1,596.6	.558	22.0	122.75	24,300
West Virginia	13,720	502-0	69-6	24-1	1,596.6	.558	22.15	127.45	25,726
Maryland	13,720	502-0	69-6	24-1	1,596.6	.558	22.4	127.8	27,570
California	13,750	502-0	69-6	24-1	1,596.6	.558	22.2	124.6	28,943
South Dakota	13,750	502-0	69-6	24-1	1,596.6	.558	22.0	122.1	24,100
Washington	14,495	502-0	72-10½	25-0	1,732	.554	22.0	121.6	25,500
Tennessee	14,480	502-0	72-10½	24-11½	1,732	.554	22.0	126.0	26,000
North Carolina	14,518	502-0	72-10½	25-0	1,732	.554	22.0	119.0	27,100
Montana	14,518	502-0	72-10½	25-0	1,732	.554	22.0	120.2	25,800
Columbia, Wing	7,375	411-7	58-2	22-5½	1,120	.491	22.8	133.5	12,166
Columbia, Center	7,375	411-7	58-2	22-5½	1,120	.491	22.8	127.7	5,286
Minneapolis, W.	7,387	411-7	58-2	22-6	1,125	.491	23.07	132.5	13,148
Minneapolis, C.	7,387	411-7	58-2	22-6	1,125	.491	23.07	132.2	7,219
St. Louis	9,665	424-0	66-0	22-6	1,312	.539	22.0	149.3	26,100
Charleston, 1st screw	9,710	424-0	66-0	22-6	1,312	.539	22.0	142.8	26,700

Tip, and C=Broader at Tip than Standard.

Characteristics of propellers.										Prop. coeff.	
No.	No. bldg.	Diameter.	Pitch.	P./D.	P.A./D.A.	Appar. slip.	Form of P.A.	Ind. thrust. Disc.	Ind. thrust. P.A.	Bare.	App.
		ft. ins.	*ft. ins.*			*per ct.*		*sq. in.*	*sq. in.*		
2	3	15-6	16-0	1.03	.24	24.85	B	2.85	11.48	...	...
2	3	15-6	16-0	1.03	.24	23.0	B	3.1	12.94	52.2	60.5
2	3	15-0	15-6	1.03	.312	14.33	B	3.55	11.33	55.1	64.2
2	3	16-6	19-6	1.18	.269	20.93	B	2.85	10.58	55.8	69.5
2	4	14-6½	16-1	1.108	.345	11.51	...	3.41	9.59	...	...
2	3	14-6½	17-4	1.19	.320	16.9	A	2.67	8.33	...	...
2	3	16-9	17-3	1.03	.313	12.93	A	3.10	9.89	53.1	65.1
2	3	16-9	17-3	1.03	.313	13.11	A	3.08	9.83	53.1	65.1
2	3	16-9	17-3	1.03	.313	13.12	A	3.24	10.30	...	71.6?
2	3	16-0	17-6	1.094	.309	14.8	B	3.22	10.42	...	71.1
2	3	15-6	17-3	1.13	.302	16.68	A	3.58	11.87	...	68.2
2	3	16-0	18-0	1.25	.323	20.7	A	3.78	11.74	65.1	71.2
2	3	17-0	18-2½	1.065	.276	13.3	A	3.70	13.42	62.0	72.3
2	3	16-6	17-3	1.045	.361	15.55	B	4.12	11.41	56.4	62.7
2	3	18-0	19-2	1.065	.303	16.16	AC	4.14	13.65	54.9	63.0
2	3	17-9	18-8	1.052	.332	17.76	C	4.22	12.7	53.9	61.6
2	3	17-9	18-8	1.052	.332	17.0	C	4.22	12.7	54.7	62.7
2	3	17-9 1/100	19-0.41	1.07	.287	16.2	B	4.16	14.5	56.0	64.2
2	3	17-3	18-3	1.06	.303	18.4	A	4.6	15.17	51.4	60.0
2	3	17-9	17-7 ¼	.99	.340	14.9	B	3.7	10.85	52.2	61.8
2	3	17-3	18-0	1.043	.308	14.9	A	3.58	11.64	51.0	61.2
2	3	17-3	18-0.7	1.043	.308	13.0	A	4.13	13.4	52.1	62.4
2	3	17-3	18-0	1.043	.308	15.0	A	4.12	13.3	50.6	59.6
2	3	17-9	18-0	1.014	.313	12.6	C	3.9	12.45	49.4	58.2
2	3	17-4	17-6	1.009	.345	13.7	A	4.4	12.75	42.0	50.4
2	3	16-6	19-0	1.152	.327	18.6	AC	4.41	13.48	53.4	...
2	3	17-6½	18-0	1.026	.315	12.2	A	4.1	13.1	62.4?	75.8?
2	3	18-3	19-9	1.082	.328	13.3	AC	4.64	14.15	55.0	64.8
2	3	16-6	17-0	1.03	.289	15.0	A	3.35	11.6	46.5	53.8
2	3	16-0	16-9	1.047	.317	17.3	A	3.43	10.8	47.5	55.0
2	3	16-0	20-0	1.425	.280	24.48	B	3.41	12.13	...	62.2
2	3	16-6	20-9	1.21	.291	18.67	B	3.58	12.29	...	69.6
2	3	18-0	22-0	1.22	.290	18.6	A	4.04	13.9	56.0	63.0
2	3	18-0	22-0	1.22	.290	17.14	A	4.04	13.87	57.0	64.3
2	3	17-3	22-0.5	1.275	.282	19.95	A	4.35	15.4	53.6	61.8
2	3	17-3	22-6	1.304	.281	21.07	A	4.57	16.27	52.9	60.3
2	3	18-0	21-6	1.19	.308	16.01	A	4.86	15.78	41.8	55.0
2	3	18-0	21-6	1.19	.308	15.7	A	4.42	14.31	51.2	59.6
2	3	18-0	21-9	1.21	.318	15.4	AC	4.35	13.71	50.9	57.5
2	3	18-0	22-0	1.22	.290	20.21	A	4.24	14.53	48.5	56.0
2	3	18-0	22-6	1.25	.306	16.13	B	4.37	14.26	46.3	57.5
2	3	18-0	21-9	1.21	.310	14.9	B	4.5	14.5	49.0	59.3
2	3	15-0	21-6	1.45	.234	19.47	B	2.75	11.77	...	...
1	3	14-0	21-6	1.55	.261	15.85	B	2.87	11.0	...	...
2	3	15-0	22-0	1.46	.234	19.8	B	2.93	12.54	...	...
1	3	14-0	21-6	1.55	.261	17.73	B	3.78	14.48	...	...
2	3	18-0	19-5½	1.083	.302	23.6	AC	4.09	13.54	49.8	55.7
2	3	16-3	20-6	1.26	.311	25.5	AB	4.94	15.91	49.4	56.0

Form of P.A. A=Standard ; B=Narrower at

Vessels.	Characteristics of hull.						Performance.		
	Displacement.	Length on L.W.L.	Greatest immersed beam.	Mean draught.	Area mid-ship section.	Block coeff.	Speed.	Revolutions per minute.	I.H.P. on all screws.
	tons.	*ft. ins.*	*ft. ins.*	*ft. ins.*	*sq. ft.*		*knots.*		
Charleston, 2d screw	9,740	424-0	66-0	22-6⅞	1,312	.539	21.3	138.0	23.000
Milwaukee	9,700	424-0	66-0	22-6	1,312	.539	22.0	138.25	22,850
Baltimore	4,391	315-0	48-6	19-6¼	816.6	.515	20.8	120.5	9,248
Philadelphia	4,325	327-6	48-6⅛	19-2½	815	.515	19.67	119.5	8,533
Newark	3,980	310-10	49-2	18-3½	775	.510	19.0	127.0	8,512
San Francisco	4,088	310.0	49-2	18-9	770	.509	19.51	124.8	9.581
Cincinnati	3,000	300-0	42-0	18-0	689	.528	19.27	140.8	6.714
Raleigh	3,574	300-0	42-0	18-9½	689	.528	19.27	140.8	6,714
Olympia	5,586	340-0	53-6	20-8½	970	.517	21.69	139.2	16,850
Montgomery	2,091	257-0	37-0	14-6¼	471	.592	19.05	180.2	5.484
Detroit	2,068	257-0	37-0	14-5⅝	467.9	.592	18.71	170.0	5,154
Marblehead	2,073	257-0	37-0	14-4⅝	464	.592	18.4	170.0	4.863
Mars	11,282	387-0	53-1¼	24-8½	...	.799	13.2	98.5	4,150
Cyclops	19,270	528-0	65-2¾	27-6	1,750	.726	15.5	98.2	8,400
Birmingham	3,750	420-0	47-1.1	16-9½	528	.405	24.0	191.0	14,700
Denver	3,200	292-0	44-0	15-9	609	.553	16.75	194.25	6,073
Cleveland	3,200	292-0	44-0	15-9	609	.553	16.5	185.0	4,600
Chattanooga	3,156	292-0	44-0	15-7	609	.553	16.5	179.5	4,950
Des Moines	3,200	292-0	44-0	15-9	609	.553	16.65	188.24	5,059
Galveston	3,250	292-0	44-0	15-11	609	.553	16.5	189.0	5.175
Tacoma	3,250	292-0	44-0	15-11	6.9	.553	16.5	196.0	5.000
Machias	1,067.5	204-0	32-1½	11-9½	324.6	.504	15.46	216.4	1,795
Castine	1,067.5	204-0	32-1½	11-9½	324.6	.504	16.03	218.8	2.128
Nashville	1,364	220-0	38-0	10-11⅝	370	.518	16.29	308.4	2,488
Helena	1,340	250-9	40-0	8-7⅝	327.1	.540	15.5	279.9	1,953
Wilmington	1,342	250-9	40-0	8-7¾	329.5	.540	15.08	273.0	1,862
Annapolis	943	168-0	36-0	11-5½	333	.482	13.17	146.7	1,208
Vicksburg	964	168-0	36-0	11-7¾	342	.482	12.67	146.5	1,084
Newport	972	168-0	36-0	11-8½	344	.482	12.33	142.6	976
Princeton	1,000	168-0	36-0	12-6½	372	.482	11.0	122.7	923
Wheeling	1,000	174-0	34-0	12-0⅝	352	.508	12.88	231.0	1,050
Marietta	991	174-0	34-0	11-11½	349	.508	13.02	231.8	1,024
Bancroft	832	187-6	32-0	11-5¾	277	.425	14.37	222.4	1,184
Paducah	1,065	174-0	35-0	12-3	352	.520	12.823	229.4	1,217
Dubuque	1,065	174-0	35-0	12-3	352	.520	12.9	230.01	1,183
Monterey	4,000	256-0	59-1⅛	14-5	772	.643	13.6	161.5	4,987
Ozark	3,275	252-0	50-0	12-7	597	.719	12.71	214.5	2,190
Tallahassee	3,275	252-0	50-0	12-7	597	.719	12.4	199.61	2,317
Tonopah	3,275	252-0	50-0	12-7	597	.719	13.04	187.0	1,940
Cheyenne, 1st screw	3,275	252-0	50-0	12-7	597	.719	11.5	214.7	2,811
Cheyenne, 2d screw	3,250	252-0	50-0	12-7	597	.719	11.0	195.0	2,500
Cheyenne, 3d screw	3,250	252-0	50-0	12-7	597	.719	12.5	196.0	2,550
Katahdin	2,125	250-2	41-8	14-10½	478.2	.461	16.09	139.54	4,904
Perry	470	245-0	23-6½	6-8½	126.7	.410	28.5	373.0	9,200
Paul Jones	475	245-0	23-6½	6-9	126.7	.410	28.5	334.0*	7.500
Preble	480	245-0	23-6½	7-2	126.7	.410	28.5	349.0	7,500
Bainbridge	460	240-6	23-0½	7-1⅝	112.5	.430	28.5	335.0	8,196
Barry	462	240-6	23-0½	7-1⅝	112.5	.430	28.13	324.0	7.950
Chauncey	463	240-6	23-0½	7-1⅝	112.5	.430	28.64	327.6	8,130
Stewart	435.0	245-0	23-7½	7-0½	112.5	.410	29.69	335.0	...

Tip, and C=Broader at Tip than Standard.

Characteristics of propellers.										Prop. coeff.	
No.	No. bldg.	Diameter.	Pitch.	P. / D.	P.A. / D.A.	Appar. slip.	Form of P. A.	Ind. thrust. Disc.	Ind. thrust. P.A.	Bare.	App.
		ft. ins.	*ft. ins.*			*per ct.*		*sq. in.*	*sq. in.*		
2	3	17-6	20-6	1.17	.320	23.56	AB	3.88	12.12	50.9	58.7
2	3	17-0	19-3	1.13	.329	11.6	AC	4.57	13.89	56.2	63.6
2	3	14-6	20-0	1.38	.272	12.6	B	2.65	9.73	...	...
2	3	14-6	20-5	1.41	.265	18.2	B	2.42	9.14	...	...
2	3	14-6	18-11	1.308	.242	20.1	B	2.47	10.22	...	...
2	3	13-6	18-9	1.39	.311	14.45	B	3.08	9.89	...	...
2	3	13-6	15-6	1.15	.279	10.53	B	2.46	8.81	...	...
2	3	13-6	15-6	1.15	.279	10.53	B	2.46	8.81	...	...
2	3	14-9	19-0	1.288	.313	16.87	B	4.27	13.63	...	68.2
2	3	11-0	12-9	1.16	.243	15.49	A	2.65	10.19	...	...
2	3	11-0	13-0	1.181	.243	14.64	A	2.81	11.57	...	...
2	3	11-0	12-0	1.09	.289	12.08	A	2.77	9.6	...	...
2	4	14-9	14-3	.966	.347	4.7	A	1.98	5.71	62.0	68.2
2	3	16-6	18-0	1.09	.278	11.14	A	2.54	9.14	59.5	...
2	3	12-6	15-3.4	1.223	.336	15.1	C	4.6	13.7	51.9	56.3
2	3	11-0	11-11	1.0	.474	27.22	C	3.21	6.78	45.3	...
2	3	10-6	11-8.64	1.119	.349	23.2	A	2.9	8.30	55.5	...
2	3	10-6	11-9	1.119	.335	22.0	A	3.1	9.25	51.2	...
2	3	10-6	11-9	1.119	.335	23.54	C	3.03	9.05	46.0	...
2	3	10-6	11-9	1.119	.335	24.7	C	3.1	9.25	48.8	...
2	3	10-0	11-6	1.15	.245	34.0	AB	3.6	14.7	50.0	...
2	3	7-8	8-10	1.141	.238	12.7	B	2.38	9.99	...	...
2	3	7-8	8-9	1.137	.238	15.6	B	2.76	11.60	...	...
2	3	6-8	7-0	1.05	.295	23.5	A	3.63	12.30	...	...
2	3	7-0	7-0½	1.015	.406	20.06	AC	2.97	7.31	...	...
2	3	7-0	7-0¼	1.018	.406	21.01	AC	2.89	7.12	...	...
1	4	9-6	11-0	1.158	.338	17.3	AC	2.42	7.17	...	...
1	3	10-0	10-7½	1.063	.243	17.39	AC	2.03	8.33	...	...
1	3	10-0	10-5½	1.045	.243	16.49	AC	1.89	7.79	...	...
1	4	10-0	10-1¼	1.0104	.328	10.01	A	2.16	6.60	...	...
2	3	6-9	7-3	1.074	.390	22.2	A	2.01	5.16	...	...
2	3	6-9	7-3	1.074	.390	21.5	A	1.95	5.01	...	...
2	3	7-0	7-9	1.107	.276	15.5	B	2.04	7.38	...	...
2	3	7-3	6-10	.943	.307	17.05	C	...	...	...	...
2	3	7-3	6-10	.943	.307	17.05	C	...	...	...	...
2	3	10-2	11-8	1.148	.358	26.8	B	3.81	10.61	...	...
2	3	9-0	7-3	.805	.355	9.94	A	2.17	6.11	54.3	...
2	3	8-6	8-4	.980	.397	24.29	A	2.82	7.11	...	...
2	3	9-2.7	7-9.55	.848	.314	9.53	A	2.28	7.26	72.0?	...
2	3	8-0	8-6	1.063	.423	33.8	B	2.6	6.15	36.0	...
2	3	9-0	7-6	.833	.508	23.8	C	3.1	6.10	24.0	...
2	3	9-0	8-6½	.95	.466	24.95	C	2.7	5.8	39.0	...
2	3	12-0	14-0	1.167	.251	19.9	B	2.52	10.04	...	...
2	3	7-5	10-10	1.46	.275	28.5	AB	6.06	22	40.2	...
2	3	7-5	10-8	1.44	.430	18.7	A	5.6	13.01	49.4	...
2	3	7-5	10-5	1.40	.358	21.2	AB	5.43	15.17	49.4	...
2	3	8-0	10-7.3	1.326	.400	20.6	A	5.4	13.5	42.5	...
2	3	8-0	11-3	1.40	.400	22.43	A	4.99	12.49	...	...
2	3	8-0	11-0¼	1.384	.400	20.57	A	5.14	12.84	...	...
2	3	8-0	10-8	1.34	.350	16.1	...	...	...	...	...

Form of P.A. A=Standard ; B=Narrower at

Vessels.	Characteristics of hull.						Performance.		
	Displacement.	Length on L.W.L.	Greatest immersed beam.	Mean draught.	Area midship section.	Block coeff	Speed.	Revolutions per minute.	I.H.P. on all screws.
	tons.	*ft. ins*	*ft. ins*	*ft. ins.*	*sq ft.*		*knots.*		
Dale	457.0	245-0	23-1½	7-1⅞	108.8	.461	28.005	331.75	...
Decatur	450.0	245-0	23-1½	6-11½	107.52	.467	28.23	331.45	...
Truxton	479.3	258-6	23-3	7-9¾	139.0	.370	29.58	328.7	7,846
Whipple	480.2	258-6	23-3	7-10	139.0	.370	28.24	316.7	7,202
Worden	482.3	258-6	23-3	7-10 7/16	139.0	.370	29.78	319.6	7,987
Hopkins	479.0	244-0	24-6	7-4½	97.8	.400	28.0	364.5	8,400
Hull	466.0	244-0	24-6	7-2½	97.8	.400	28.2	418.5	8,978
Lawrence	412.0	240-10	22-3	6-6½	110.5	.404	29.0	364.5	10,184
Macdonough	412.0	240-10	22-3	6-6½	110.5	.404	28.0	360.46	...
Ericsson	126.0	149-7	15-6	4-10⅝	51.5	.391	21.24	375.1	1,742
Cushing	105.3	137-6	14-3	4-10½	48.0	.386	22.5	370.0	1,755
Foote	138.3	160-0	16-0¾	5-3½	58.5	.377	24.53	395.7	...
Rodgers	147.0	160-0	16-0¾	5-5	64.5	.364	24.25	395.8	2,306
Winslow	138.0	160-0	16-0¾	5-3½	58.5	.370	24.82	393.92	...
Porter	165.0?	175-0	17-8½	4-9½	60.0	.399	28.6	389.5	...
Dupont	165.0?	175-0	17-8½	4-9½	60.0	.399	28.51	396.3	...
Rowan	184.25	170-0	17-0	5-11½	78.0	.372	27.07	368.0	...
Dahlgren	132.0	147-0	16-4½	4-3½	60.0	.447	30.01	314.4	...
Craven	133.0	147-0	16-4½	4-3⅝	60.0	.448	30.5	334.0	3,600
Farragut	236.3	213-6	20-7¾	6-5 1/16	82.0	369	30.62	427.6	...
Davis	126.0	146-0	15-4	5-3⅝	63.0	.371	23.4	382.0	...
Fox	123.0	146-0	15-4	5-2	63.0	.372	23.13	380.4	...
Talbot	45.75	99-6	12-6	3-9⅝	29.0	.337	21.15	437.9	...
Gwin	44.5	99-6	12-6	3-9½	29.0	.334	20.88	442.3	...
Morris	98.0	138-3	15-6	4-2½	52.2	.380	24.0	451.9	...
Mackenzie	76.0	99-3	12-9½	4-8½	38.0	.420	20.11	391.1	...
McKee	76.0	99-3	12-9½	4-8½	36.5	.420	19.82	370.0	...
Stringham	340.0	225-0	22-0	...	103.5	.418	26.0	356.0	...
Goldsborough	255.0	194-8	20-5	6-10	78.4	.448	25.2	290.0	...
Bailey	280.9	205-0	18-9½	...	72.77	.374	30.88	422.0	...
Bagley	168.0	157-0	17-7½	4-10	56.8	.478	29.15	309.12	3,920
Barney	168.0	157 0	17-7½	4-9⅝	56.8	.478	29.04	312.31	3,910
Biddle	168.0	157-0	17-7½	4-10	56.8	.478	28.51	306.7	3,650
Blakeley	192.0	175-1½	16-10⅝	...	63.4	.375	25.49	371.45	...
De Long	192.0	175-1½	16-10⅝	5-9 3/32	63.4	.375	25.52	360.4	...
Shubrick	188.5	175-0	17-6	5-9¾	72.0	.389	26.22	365.8	3,275
Stockton	197.0	175-0	17-6	5-11	74.0	.399	25.88	358.0	3,218
Thornton	174	175-0	17-6	5-9½	71	383	27.57	392.6	3,576
Wilkes	200	175-0	17-6	5-11 9/16	78	.402	25.99	396.6	...
Tingey	187	175-0	17-6	5-0⅝	71	.437	24.0	346.3	...
Nicholson	219	...	...	...	...	.475	25.0	298.0	4,150
O'Brien	219	...	...	...	...	.475	23.0	305.0	2,760
50-ft. Launch	19.77	50-0	10-0	3-11½	...	352	11.0	412.0	194.5
Drake	14,100	500-0	71-0	26-1½	...	.532	24.11	122.4	31,409
Good Hope	14,100	500-0	71-0	26-1½	...	.532	23.05	126.2	31,088
Leviathan	14,100	500-0	71-0	26-1½	...	.532	23.23	121.0	31,208
King Alfred	14,100	500-0	71-0	26-1½	...	.532	23.46	118.8	30,893

Turbine Vessels. I.H.P=S.H.P.÷.92 ; P.C.=

Tip, and C=Broader at Tip than Standard.

Characteristics of propellers.										Prop. coeff.	
No.	No. bldg.	Diameter.	Pitch.	P./D.	P.A./D.A.	Apprar. slip.	Form of P.A.	Ind. thrust. Disc.	Ind. thrust. P.A.	Bare.	App.
		ft. ins.	*ft. ins.*			*per ct.*		*sq. in.*	*sq. in.*		
2	3	7-10	10-3	1.308	.331	13.8	...	...	...	...	...
2	3	7-10	10-3	1.308	.331	16.38	...	...	...	...	...
2	3	8-0	11-3	1.406	.341	18.93	B	4.84	14.19	...	...
2	3	8-0	11-3	1.406	.341	19.72	B	4.61	13.51	...	...
2	3	8-0	11-3	1.406	.341	18.84	B	5.06	14.85	...	...
2	3	8-6	8-6.22	1.000	.388	9.82	C	5.4	13.95	...	...
2	3	7-8	8-3	1.078	.379	19.7	B	6.49	17.05	...	...
2	3	7-5	9-6	1.3	.403	15.2	C	7.8	19.35	41.5	...
2	3	7-4½	9-6	1.3	.403	17.1	C	...	...	...	...
2	4	4-10	6-4	1.31	.449	9.4	C	4.57	10.17	...	...
2	4	4-3	8-5	1.98	.402	26.8	C	4.56	11.35	...	...
2	3	5-2	7-10	1.51	.366	19 5	AB	...	...	...	...
2	3	5-3	7-10	1.49	.366	20.94	A	3.93	10.74	...	...
2	3	5-3	7-10	1.49	.366	18.16	A	...	...	...	...
2	3	5-4	8-8½	1.63	.325	14.3	C	...	...	...	...
2	3	5-4	8-8½	1.63	.325	...	C	...	...	...	...
2	4	6-0	9-0	1.5	.355	17.2	A	...	...	...	...
2	3	6-5	11-5½	1.79	.280	16.4	A	...	...	...	...
2	3	6-5	11-6½	1.8	.280	17.3	A	3.5	12.5	62.0	75.8
2	3	6-9	8-9	1.29	.251	17.25	A	...	...	...	...
2	3	4-10½	7-3	1.48	.343	14.48	B	...	...	...	...
2	3	4-10½	7-3	1.48	.343	15.02	B	...	...	...	...
1	3	4-8	6-0.6	1.29	.310	19.4	C	...	...	...	...
1	3	4-8	6-3½	1.33	.309	23.8	C	...	...	...	...
2	3	4-8	6-2⅝	1.33	.309	13.5	C	...	...	...	...
1	4	5-3	6-9	1.28	.393	22.8	B	...	...	...	...
1	4	5-3	6-7	1.25	.393	17.2	C	...	...	...	...
2	3	7-8	8-6	1.1	.379	16.0	B	...	...	...	...
2	3	7-6	10-2⅞	1.36	.656	13.6	AC	...	...	...	...
2	3	6-10	9-1½	1.33	.314	18.74	B	...	...	...	...
2	3	6-8	10-10.3	1.62	.300	12.2	A	4.01	13.38	...	...
2	3	6-8	10-10.3	1.62	.300	15 7	A	3.96	13.21	...	...
2	3	6-8	10-10.3	1.62	.300	13.2	A	3.76	12.56	...	...
2	4	5-6	9-6⅜	1.74	.308	22.02	C	...	...	...	...
2	4	5-6	9-6	1.74	.308	25.2	C	...	...	...	...
2	3	5-8	9-0	1.59	.302	20.06	AC	4.54	15.02	...	...
2	3	5-8	8-11½	1.58	.302	18.8	AC	4.58	15.16	...	...
2	3	5-8	9-0	1.59	.302	27.9	AC	4.61	15.28	61.2	...
2	3	5-6	8-8	1.58	.312	23.4	B	...	...	...	...
2	3	5-8	9-0	1.59	.310	22.7	C	...	...	...	...
2	3	6-5	10-0	1.61	.355	15.0	...	5.0	14.0	...	...
2	3	6-3	8-11½	1.43	.355	14.5	...	3.8	9.75	...	...
1	3	3-0	5-0	1.67	.460	46.0	C	3.06	6.65	40.6	...
2	3	19-0	23-0	1.21	.292	13.22	B	4.52	15.48	...	...
2	3	19-0	22-9	1.197	.216	18.6	B	4.45	20.58	...	...
2	3	19-0	23-9	1.25	.216	17 8	B	4.49	20.81	...	...
2	3	19-0	23-9	1.25	.216	15.7	B	4.52	20.91	...	...

$\frac{\text{E.H.P.}}{\text{S.H.P.}} \times .92$; I.T. $= \frac{\text{S.T.}}{.92}$. Motor Vessels.

Form of P.A. A = Standard; B = Narrower at

Vessels.	Characteristics of hull.						Performance.		
	Displacement.	Length on L.W.L.	Greatest immersed beam.	Mean draught.	Area midship section.	Block coeff.	Speed.	Revolutions per minute.	I.H.P. on all screws.
	tons.	*ft. ins.*	*ft. ins.*	*ft. ins.*	*sq. ft.*		*knots.*		
N. Dakota	20,020	510-2¼	85-2½	26-10⅝	2,244	.60	21.66	280.4	34,022
Salem 1	3,750	420-0	49-1	16-9½	565	.405	19 0	91.0	11,913
Salem 2	4,000	420-0	49-1	17-5½	565	.407	25.0	357.0	16,576
Salem 3	3,750	420-0	49-1	16-9½	565	.405	25.0	378.0	18,750
Salem 4	3,750	420-0	49-1	16-9½	565	.405	24.5	361.25	18,630
Creole 1	7,350	440-0	53-0	...	...	.635	14.75	219.8	6,893
Creole 2	7,200	440-0	53-0	...	...	.635	15.0	224.7	6,952
Creole 3	7,390	440-0	53-0	...	...	640	16.85	250.5	7,049
Creole 4	9,700	440-0	53-0	...	...	.650	17.0	281.2	10,704
Creole 5	9,700	440-0	53-0	...	...	.650	15.0	261.5	6,388
Castro	87.5	75-4	18-6	5-11½	...	.369	10.85	291.2	290
Chester, out	3,977	420-0	47-1	17-3½	565	.40	25.9	604.4	12,356
Chester, inn	3,977	420-0	47-1	17-3½	565	.40	25.9	602 45	13,708
Utah, out	...	...	...	...	...	...	...	...	...
Utah, inn	...	...	...	...	...	...	...	...	...
Perkins	761	289-0	26-0½	8-6½	145	.414	31.2	650.0	15,033
Sterett	754	289-0	26-0½	8-5 9/16	145	.414	30.368	631.19	13,901
Warrington	742	289-0	26-0½	8-4	143	.414	31.4	685.0	15,870
Mayrant	734	289-0	26-0½	8-3½	143	.414	30.219	659.38	14,282
Walke	772	289-0	26-1½	8-7	145	.414	29.783	621.54	13,667
Roe, Wing	711	289-0	26-1½	8-1½	143	.410	29.601	797.36	8,070
Roe, Center	711	289-0	26-1½	8-1½	143	.410	29.601	790.0	4,931
Terry, Wing	722.5	289-0	26-1½	8-2½	143	.410	30.241	845.48	9,286
Terry, Center	722.5	289-0	26-1½	8-2½	143	.410	30.241	811.03	5,225
Smith, Wing	716	289-0	26-0⅝	8-4½	135	.407	28.35	721.34	6,730
Smith, Center	716	289-0	26-0⅝	8-4½	135	.407	28.35	728.65	4,082
Lamson, Wing	690	289-0	26-0⅝	8-0	135	.405	28.605	753.76	7,578
Lamson, Center	690	289-0	26-0⅝	8-0	135	.405	28.605	728.96	3,942
Preston, Wing	719	289-0	26-0	8-2½	136	.399	29. 177	787.35	7.678
Preston, Center	719	289-0	26-0	8-2½	136	.399	29. 177	761.87	4,180
McCall, Wing	738	289-0	26-1½	8-4 1/32	136	.403	30.659	828.5	9,342
McCall, Center	738	289-0	26-1½	8-4 1/32	136	.403	30.659	772	4,870
Burroughs, Wing	720.5	289-0	26-1½	8-2½	136	.403	30.674	847.912	9,530
Burroughs, Center	720.5	289-0	26-1½	8-2½	136	.403	30.674	774.23	4,877
Ammen, Wing	735.9	289-4⅜	25-11	8-3	143.5	.412	30.483	843	10,165
Ammen, Center	735.9	289-4⅜	25-11	8-3	143.5	.412	30.483	782	5,053
Flusser, Wing	686	289-0	26-0	7-11½	135	.405	30.41	806.76	8,529
Flusser, Center	686	289-0	26-0	7-11½	135	.405	30.41	790.47	4,016
Reid, Wing	690	289-0	26-0	8-0	135	.405	31.82	850.7	8,862
Reid, Center	690	289-0	26-0	8-0	135	.405	31.82	826.5	4,640
Paulding, Wing	711	289-0	26-0	8-1⅛	143	.405	32.801	930.66	12,862
Paulding, Center	711	289-0	26-0	8-1⅛	143	.405	32.801	847.7	5,619
Drayton, Wing	721	289-0	26-0½	8-2 9/16	143	.410	30.83	860.78	11,020
Drayton, Center	721	289-0	26-0½	8-2 9/16	143	.410	30.83	812.05	5,848
Trippe, Wing	733	289-0	26-0½	8-3½	143	.410	30.891	886.18	10,309
Trippe, Center	733	289-0	26-0½	8-3½	143	.410	30.891	834.97	5.972
Monaghan, Wing	734.8	289-0	26-1½	8-4	143	.410	30.449	835.5	8,934
Monaghan, Center	734.8	289-0	26-1½	8-4	143	.410	30.449	778.71	4,556

Tip, and C = Broader at Tip than Standard.

Characteristics of propellers.										Prop. coeff.	
No.	No. bldg.	Diameter.	Pitch.	P./D.	P.A./D.A.	Appar. slip.	Form of P.A.	Ind. thrust. Disc.	Ind. thrust. P.A.	Bare.	App.
		ft. ins.	*ft. ins.*			*per ct.*		*sq. in.*	*sq. in.*		
2	3	13-0	10-4	.795	.482	24.33	B	10.13	21.02	45.36	53.60
2	3	10-6	7-6	.714	.334	12.0	Ac	7.12	21.3	27.2	30.27
2	3	9-6	8-8	.913	.543	19.5	A	9.0	16.56	52.6	59.62
2	3	9-0$\frac{1}{2}$	8-3$\frac{5}{16}$	.92	.525	22.8	B	10.3	19.62	48.8	55.84
2	4	9-0	8-6	.944	.642	19.4	A	10.98	17.10	44.5	50.6
2	4	11-6	7-2	.677	.249	5.5	B	5.7	22.82	28.24	...
2	4	11-6	6-6	.565	.250	1.5	B	6.2	24.8	29.81	...
2	4	10-9	8-2	.760	.301	16.8	Ac	4.35	14.45	43.15	...
2	3	9-6$\frac{1}{2}$	8-4$\frac{3}{8}$	.874	.597	27.8	A	7.23	12.12	42.87	...
2	3	9-9	7-1$\frac{1}{2}$	.729	.673	18.0	...	5.28	7.85	44.62	...
1	3	5-0	4-10	.967	.282	21.88	B	2.41	8.54	...	...
2	3	6-0	6-0	1.0	.60	27.6	Ac	13.81	23.0	42.68	47.82
2	3	6-0	6-0	1.0	.60	27.4	Ac	15.37	25.56		
2	3	9-2	8-6	.9273	.56	...	Ac	...	...	...	...
2	3	9-2	8-6	.9273	.56	...	Ac	...	...	...	...
2	3	6-7	6-4$\frac{3}{4}$	.967	.564	20.10	B	12.31	21.83	43.5	47.2
2	3	6-6$\frac{1}{2}$	6-3$\frac{3}{8}$	.960	.538	22.60	B	11.65	21.65	44.9	48.2
2	3	6-8	6-2.08	.925	.584	22.89	A	12.32	21.10	42.8	46.6
2	3	6-8	6-2.17	.927	.584	24.87	A	11.49	19.66	43.4	47.6
2	3	6-6$\frac{1}{2}$	6-4	.968	.542	23.33	B	11.94	22.03	42.8	46.8
2	3	5-3	4-10	.921	.60	22.16	A	11.08	18.47	43.9	48.4
1	3	5-3	4-10	.921	.60	21.47	A	13.38	22.3	...	...
2	3	5-3	4-10	.921	.60	24.97	A	12.03	20 05	42.04	46.17
1	3	5-3	4-10	.921	.60	21.77	A	14.11	23.52	...	...
2	3	4-10	5-0	1.034	.62	20.35	Ac	11.65	18.79	46.5	53.9
1	3	4-10	5-0	1.034	.620	21.15	Ac	13.99	22.57		
2	3	4-10	5-0	1.034	.620	23.12	Ac	12.56	20.26	43.4	47.7
1	3	4-10	5-0	1.034	.620	20.51	Ac	13.51	21.79		
2	3	5-0	4-9	.95	.625	20.95	A	11.95	19.12	46.4	50.2
1	3	5-0	4-9	.95	.625	18.30	A	13.44	21.50		
2	3	5-3$\frac{1}{2}$	4-10	.921	.625	22.38	A	12.26	19.62	43.8	47.9
1	3	5-3$\frac{1}{2}$	4-10	.921	.625	16.71	A	13.72	21.95		
2	3	5-3$\frac{1}{2}$	4-10	.917	.625	24.15	A	12.29	19.67	43.7	47.8
1	3	5-3$\frac{1}{2}$	4-10	.917	.625	16.94	A	13.78	22.05		
2	3	5-4	4-10	.906	.609	24.36	A	12.80	21.00	40.5	44.3
1	3	5-4	4-10	.906	.609	18.27	A	13.71	22.51		
2	3	5-3	4-10	.921	.60	20.97	A	11.58	19.3	49.4	53.4*
1	3	5-3	4-10	.921	.60	19.34	A	11.13	18.55		
2	3	5-3	4-10	.921	.60	21.55	A	11.46	19.10	51.9	56.4*
1	3	5-3	4-10	.921	.60	19.20	A	12.29	20.49		
2	3	5-3	4-10	.921	.60	26.10	A	15.13	25.22	41.2	45.0
1	3	5-3	4-10	.921	.60	18.877	A	14.52	24.20		
2	3	5-3	4-10	.921	.60	24.9	A	14.02	23.37	38.0	41.5
1	3	5-3	4-10	.921	.60	20.4	A	15.73	26.21		
2	3	5-3	4-10	.921	.60	26.87	A	12.74	21.23	39.6	43.0
1	3	5-3	4-10	.921	.60	22.43	A	15.66	26.10		
2	3	5-3	4-10	.921	.60	23.59	A	11.72	19.55	45.66	50.04
1	3	5-3	4-10	.921	.60	18.02	A	12.82	21.37		

* Deadwood cut away. Middle screw nearly as clear as wing ones.

Charts of Design.

Following this page are the Charts of basic data, Plates 1 to 3, inclusive, and the Charts of Design 4 to 22, inclusive, which are described in the preceding pages, and which it is trusted will be found to be of value in the actual work of design.

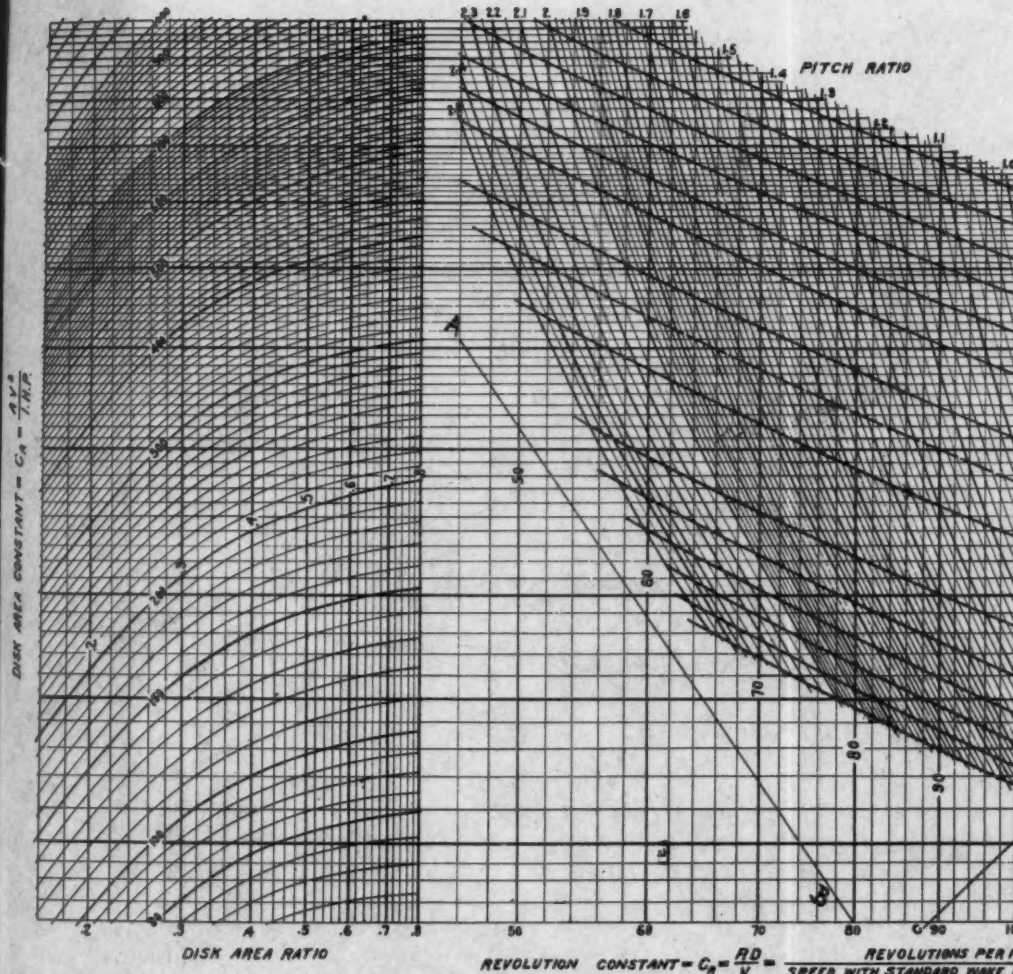
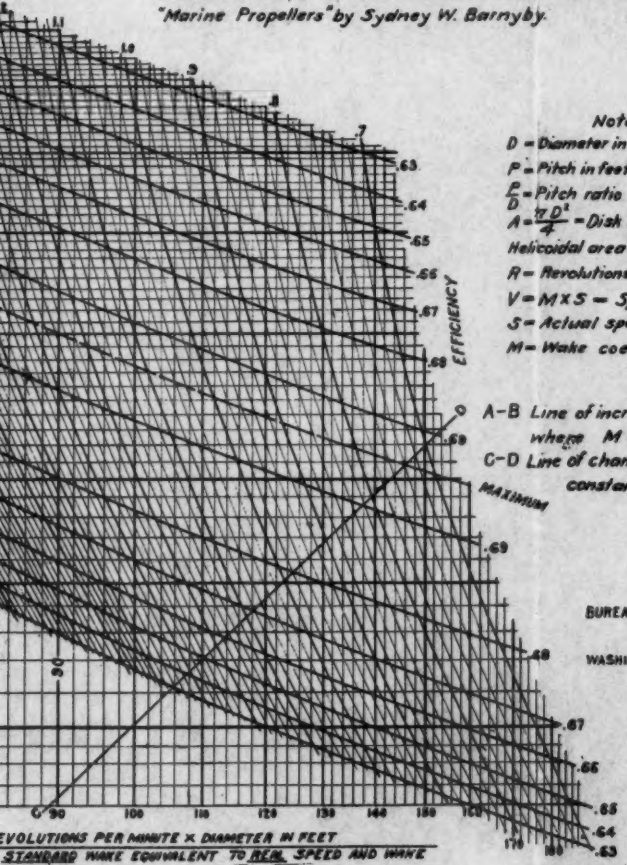

PITCH RATIO
DISK AREA CONSTANT = $C_A = \frac{AV^2}{THP}$
DISK AREA RATIO
REVOLUTION CONSTANT = $C_n = \frac{ND}{V}$ = REVOLUTIONS PER
SPEED WITH STANDARD WAKE
A

Chart showing the relation between Diameter, Pitch, Helecoidal Area, Indicated Horsepower, Revolutions, Speed and Efficiency. Based on experiments with model propellers by R.E. Froude (Trans. Inst. Naval Architects, XXVII P. 250 and on a discussion of the same in "Marine Propellers" by Sydney W. Barnyby.



Notation.

D = Diameter in feet.
P = Pitch in feet.
$\frac{P}{D}$ = Pitch ratio
$A = \frac{\pi D^2}{4}$ = Disk area
Helicoidal area ÷ A = Disk area ratio
R = Revolutions per minute
V = M x S = Speed in knots
S = Actual speed of vessel
M = Wake coefficient

A-B Line of increasing values of M where M x S = V
C-D Line of change for varying diameter constant $\frac{H.A.}{D.A.}$

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NAVY DEPARTMENT.
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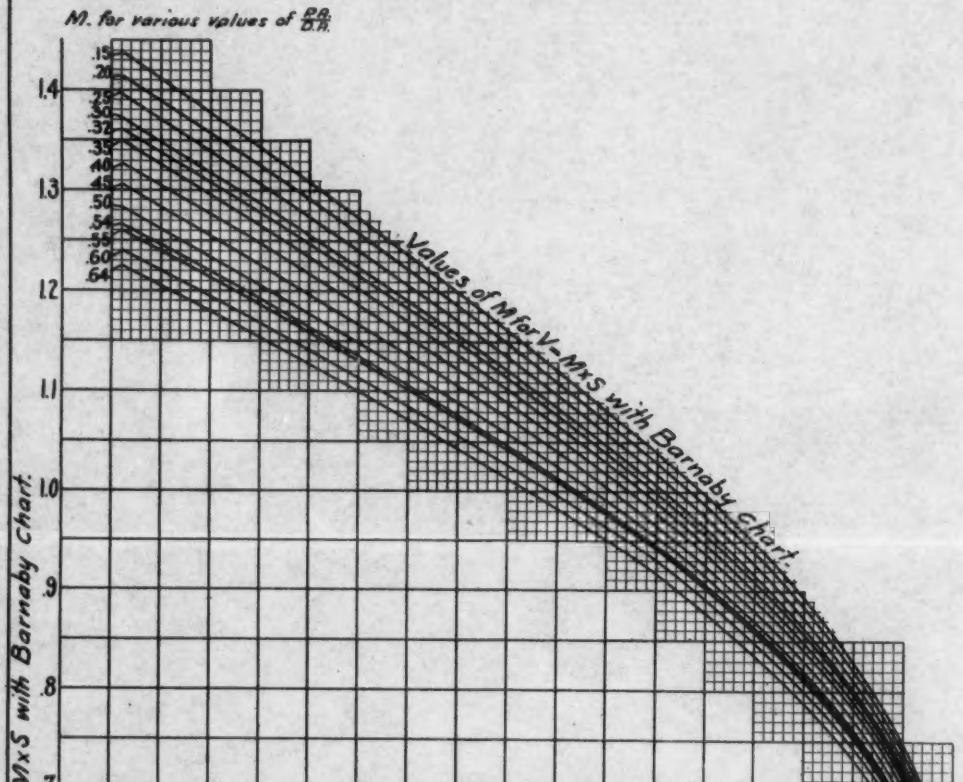
Plate 1.


M. for various values of $\frac{D_2}{D_1}$
.15
.20
.25
.30
.32
.35
.40
.45
.50
.54
.55
.60
.64
1.4
1.3
1.2
1.1
1.0
.9
.8
Values of M for V-MxS with Barnaby Chart.
V-MxS with Barnaby Chart.

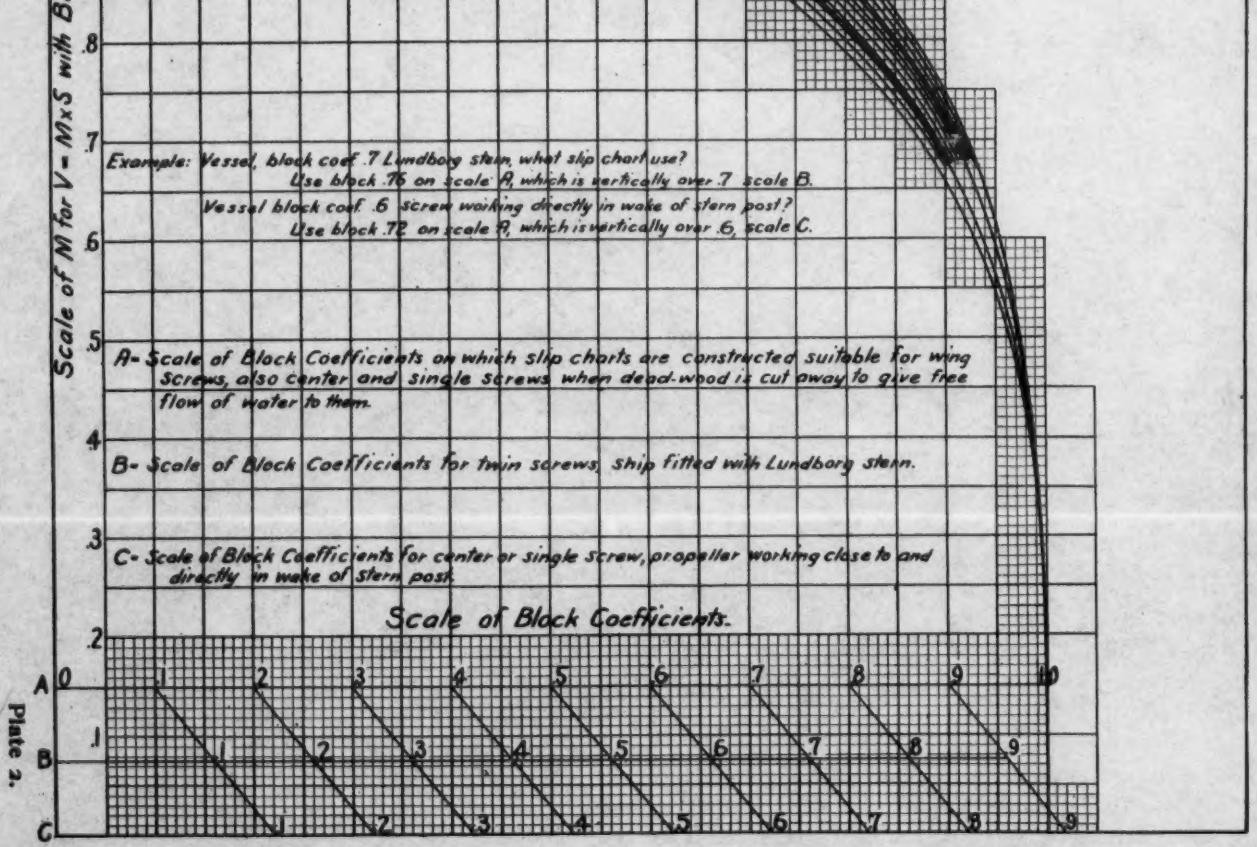
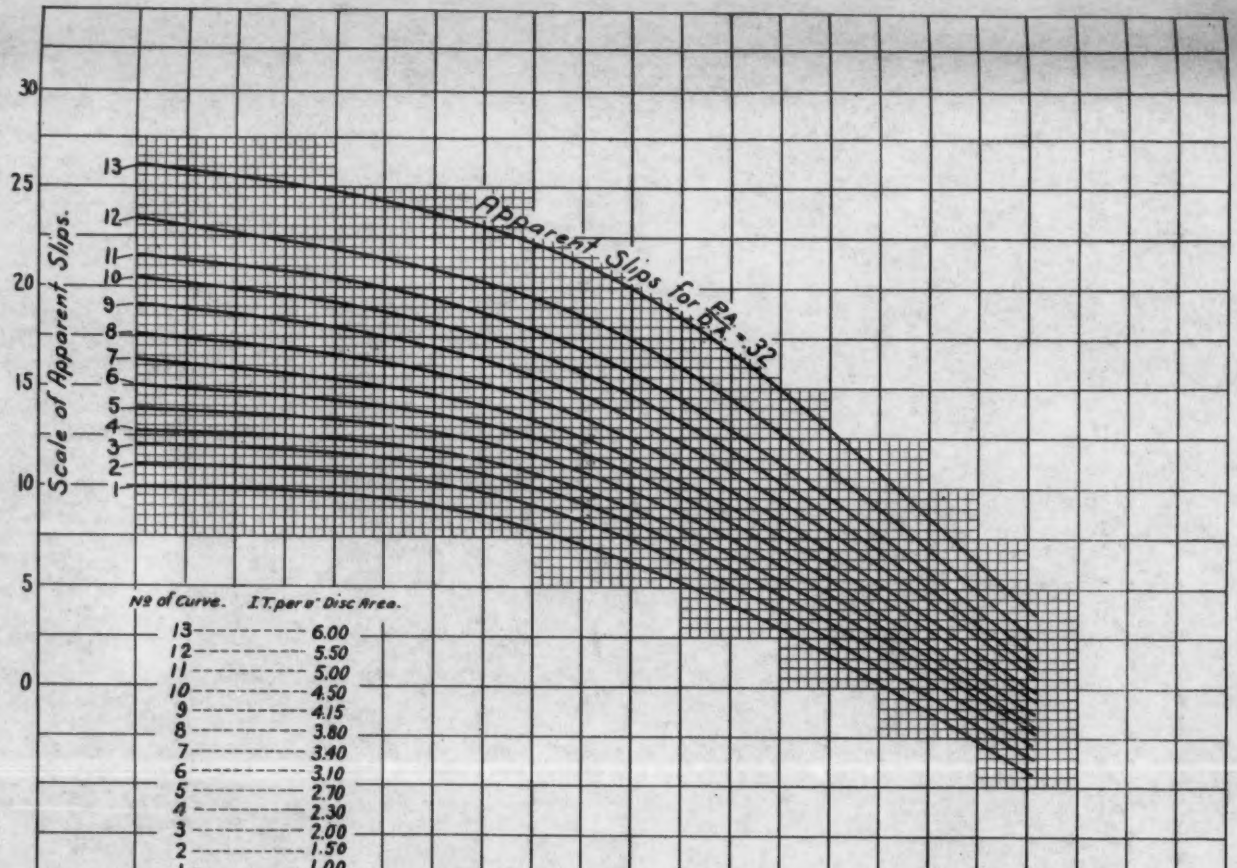

Scale of M for V = M x S with B
.8
.7
.6
.5
.4
.3
.2
Example: Vessel, block coef .7 Lundborg stern, what slip chart use?
Use block .76 on scale A, which is vertically over .7 scale B.
Vessel block coef .6 screw working directly in wake of stern post?
Use block .72 on scale A, which is vertically over .6, scale C.
A- Scale of Block Coefficients on which slip charts are constructed suitable for wing screws, also center and single screws when dead-wood is cut away to give free flow of water to them.
B- Scale of Block Coefficients for twin screws, ship fitted with Lundborg stern.
C- Scale of Block Coefficients for center or single screw, propeller working close to and directly in wake of stern post.
Scale of Block Coefficients.
A
B
C
0
1
2
3
4
5
6
7
8
9
10

Plate 2.


Apparent Slips for BA = .32 D.T.
Scale of Apparent Slips.
30
25
20
15
10
5
0
13
12
11
10
9
8
7
6
5
4
3
2
1
Nº of Curve. I.T. per 0" Disc Area.
13 ---------- 6.00
12 ---------- 5.50
11 ---------- 5.00
10 ---------- 4.50
9 ---------- 4.15
8 ---------- 3.80
7 ---------- 3.40
6 ---------- 3.10
5 ---------- 2.70
4 ---------- 2.30
3 ---------- 2.00
2 ---------- 1.50

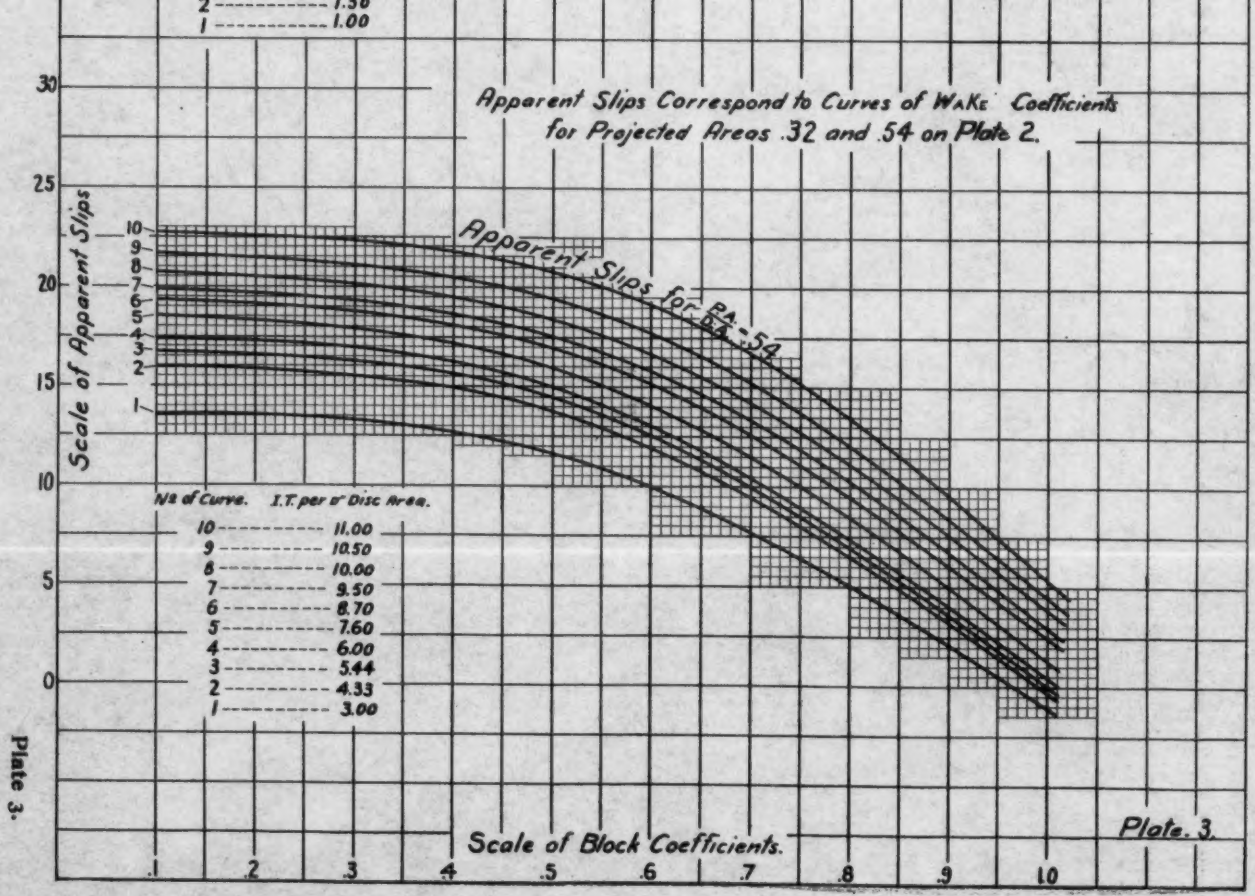
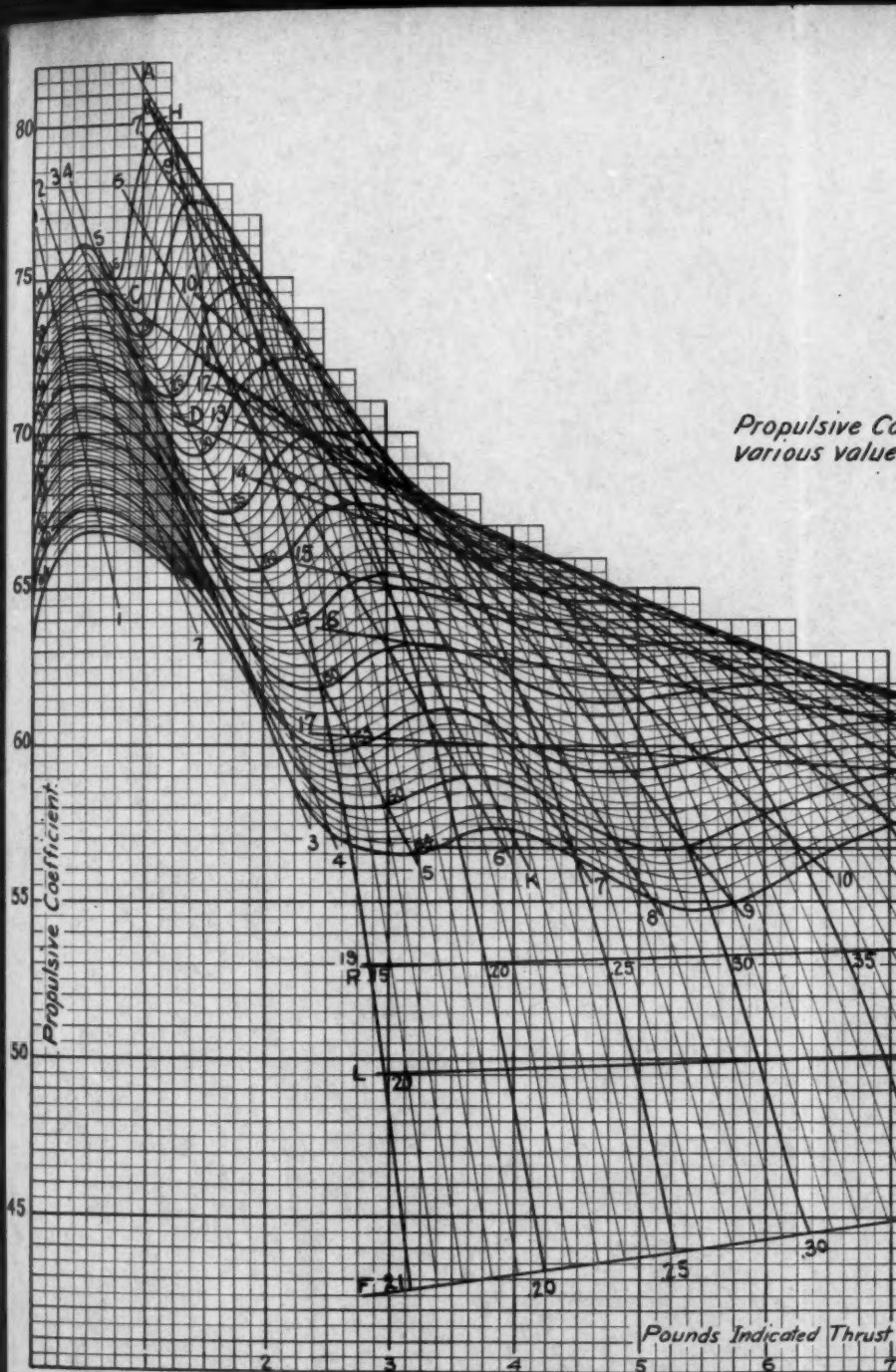
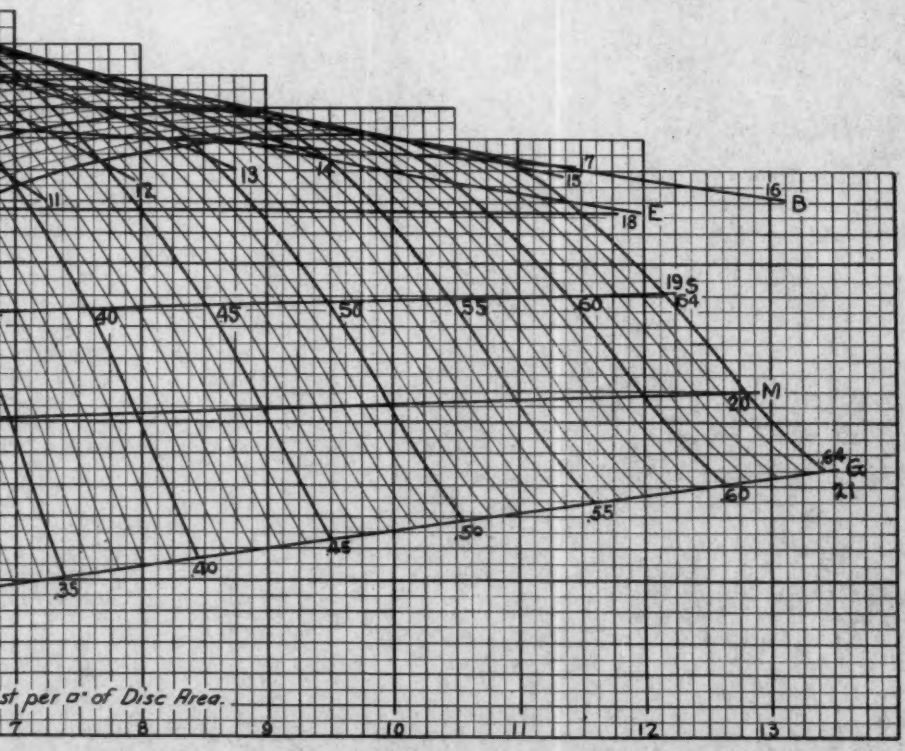

2 ---------- 1.30
1 ---------- 1.00
Apparent Slips Correspond to Curves of Wake Coefficients for Projected Areas .32 and .54 on Plate 2.
Apparent Slips for P.A. .54
Scale of Apparent Slips
30
25
20
15
10
5
0
10
9
8
7
6
5
4
3
2
1
No of Curve. I.T. per sq Disc Area.
10 ---------- 11.00
9 ---------- 10.50
8 ---------- 10.00
7 ---------- 9.50
6 ---------- 8.70
5 ---------- 7.60
4 ---------- 6.00
3 ---------- 5.44
2 ---------- 4.33
1 ---------- 3.00
Scale of Block Coefficients.
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
Plate. 3.

Plate 3.


Propulsive Co
various value
Propulsive Coefficient
Pounds Indicated Thrust
80
75
70
65
60
55
50
45
2
3
4
5
6
7
A
B
C
D
E
F
G
H
K
L
R
10
15
20
25
30
35

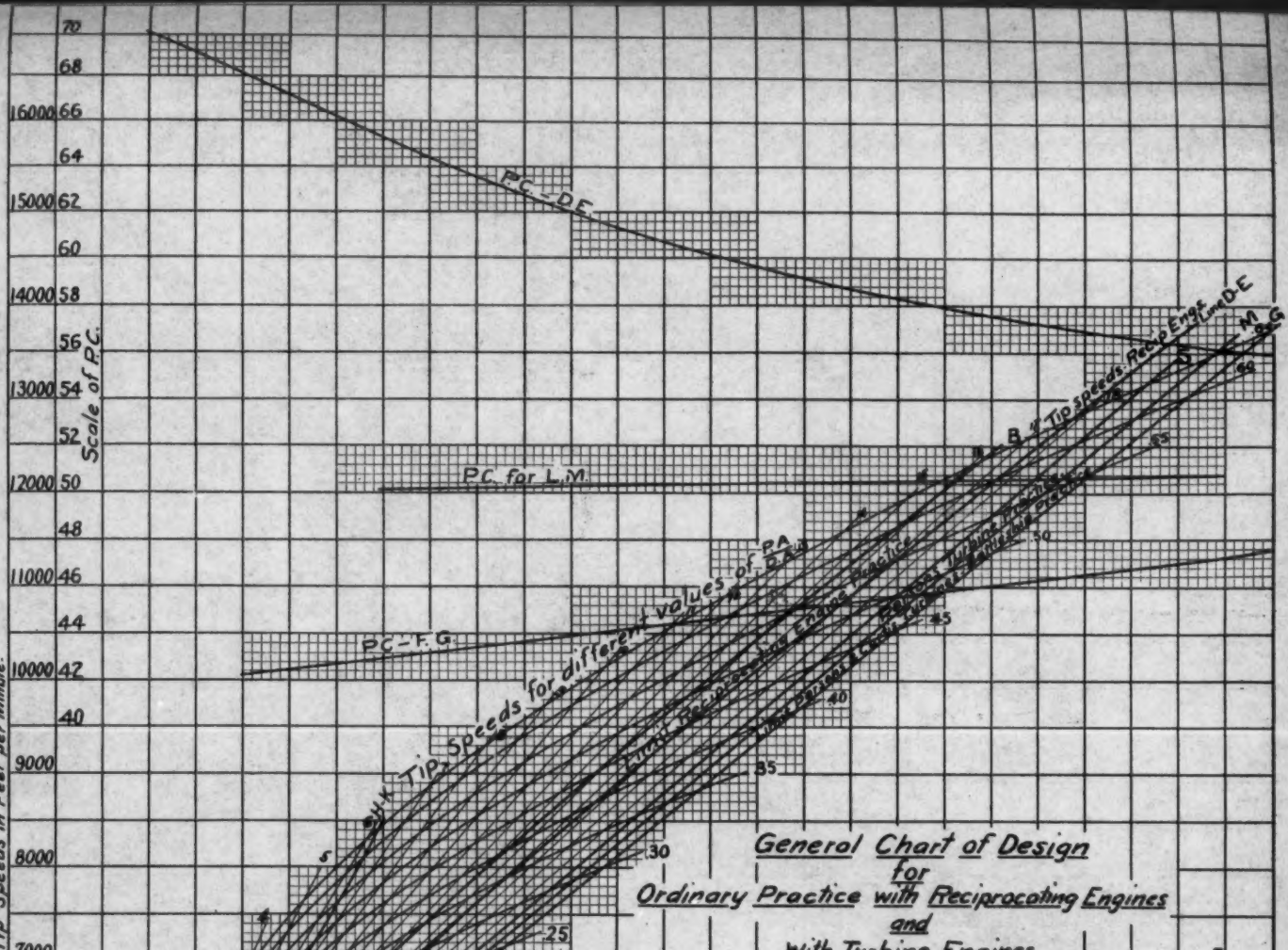
Coefficient for
ues of $\frac{P.P.}{D.P.}$

Lines numbered 1-1 to 21-21 are lines of
Equal Thrust per □" projected area.
A-B - Line of First Critical.
C-B - Line of Critical Thrusts.
D-E - Safe Line of Thrusts.
H-K - Line of Second Max. P.C.
R.S Lower Limit of Design - Turbines.
F-G - Max. Limit of Design - Turbines.



st per □" of Disc Area.

Plate 4.


Scale of P.C.
Tip Speeds in Feet per Minute.
P.C. for L.M.
P.C. - D.E.
P.C. - F.G.
Tip Speeds for different values of P.A.
Tip Speeds, Recip. Engs.
Tip Speeds, Turbine Engines
Limit, Recip. Engines
Limit, Turbine Engines
General Chart of Design
For
Ordinary Practice with Reciprocating Engines
and
with Turbine Engines

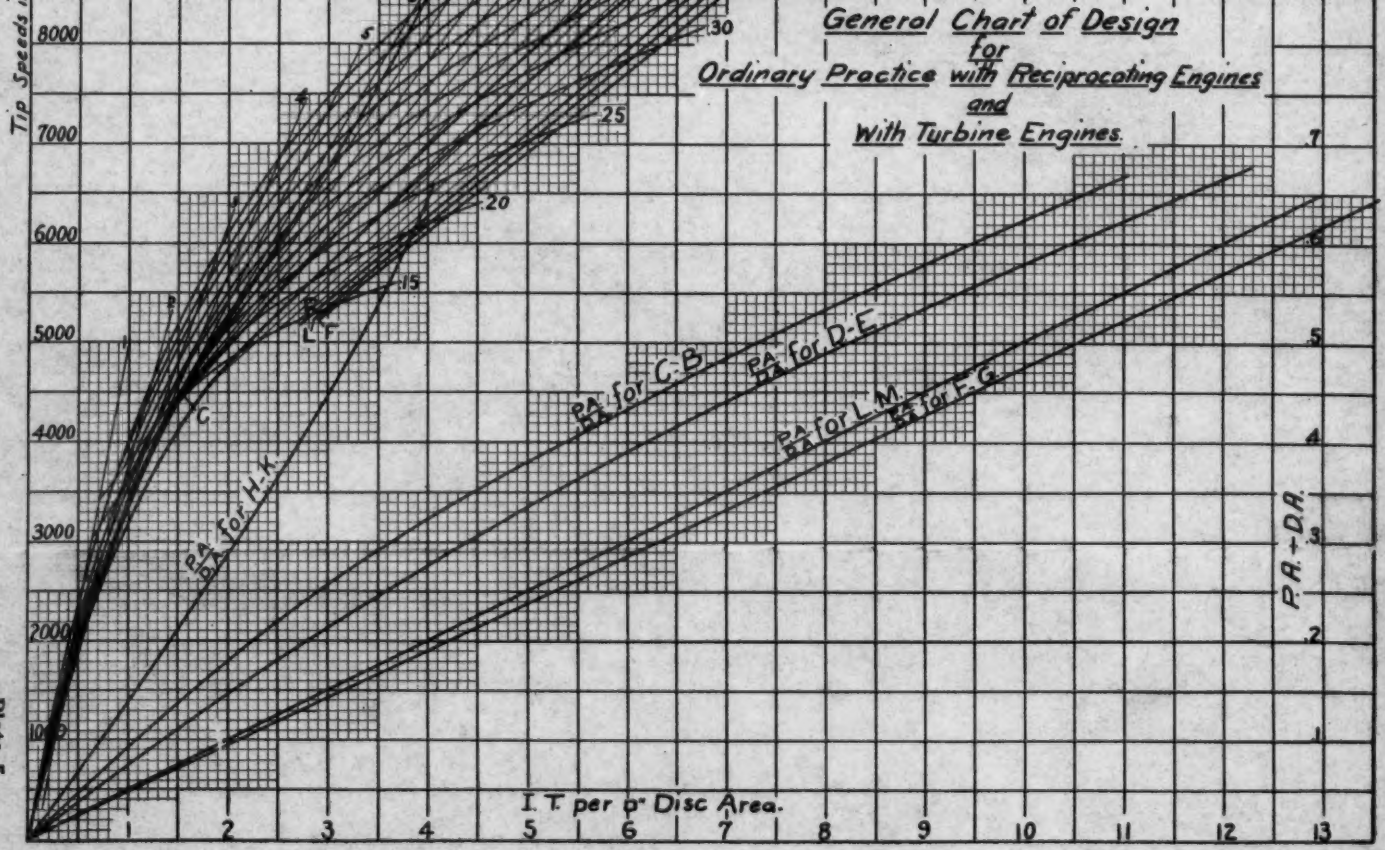
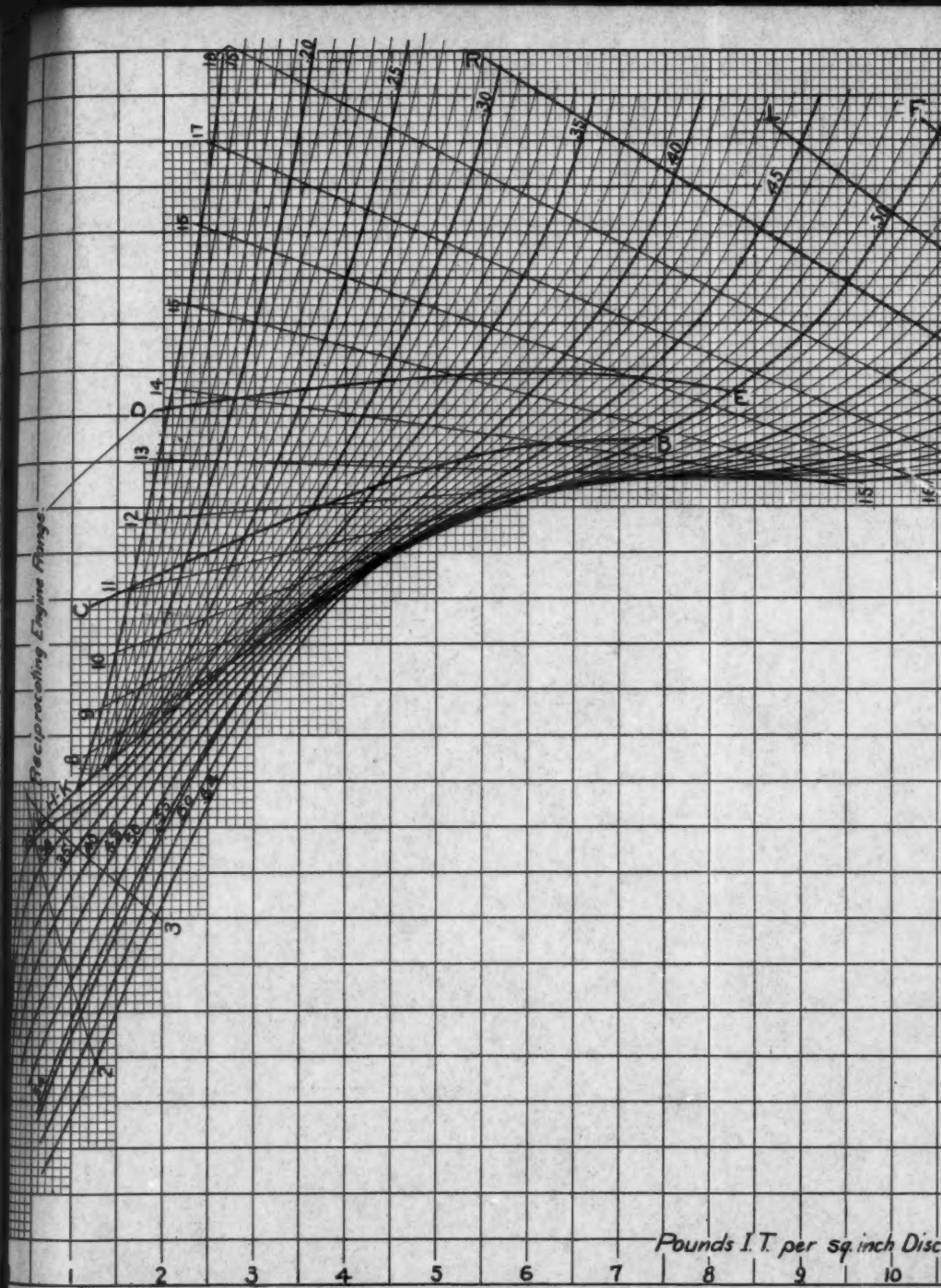
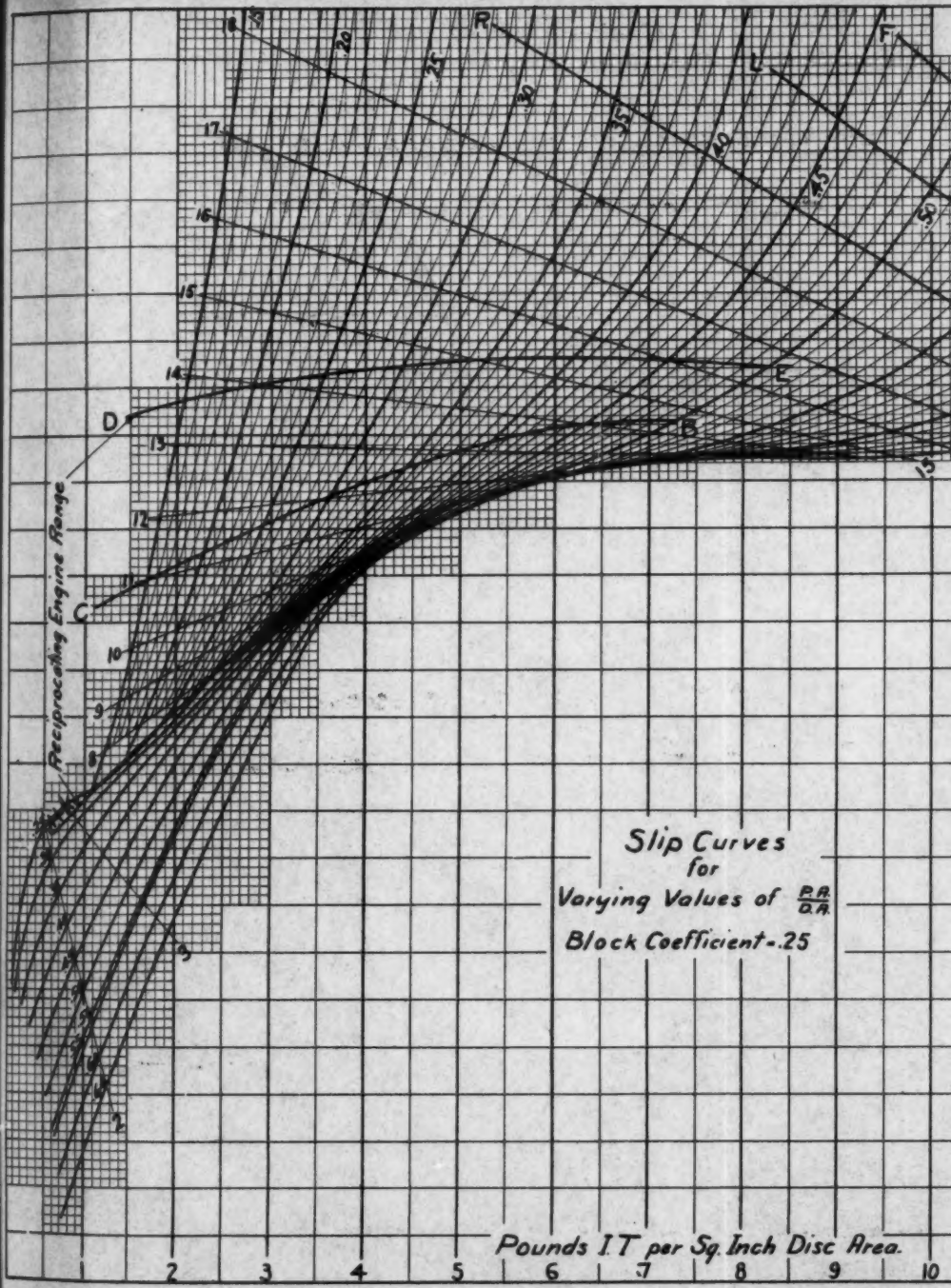

General Chart of Design
for
Ordinary Practice with Reciprocating Engines
and
With Turbine Engines.
Tip Speeds
8000
7000
6000
5000
4000
3000
2000
1000
5
4
30
25
20
15
P
C
L
F
P.A. for H.K.
P.A. for C.B.
P.A. for D.E.
P.A. for L.M.
P.A. for F.G.
P.A. ÷ D.A.
.7
.6
.5
.4
.3
.2
.1
I.T. per □" Disc Area.
1
2
3
4
5
6
7
8
9
10
11
12
13

Plate 5.


Reciprocating Engine Rings
Pounds I.T. per sq. inch Disc
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
25
30
35
40
45
50
C
D
E
F
R

Slip Curves
for
Varying Values of P.R./D.R.
Block Coefficient = .20
Apparent Slip in Percent.
sq. inch Disc Area
Max. Turbine Limit
Turbine limit
S
M
G
50
55
60
64

Plate 6.


Slip Curves
for
Varying Values of P.R. / D.R.
Block Coefficient - .25
Reciprocating Engine Range
Pounds I.T. per Sq. Inch Disc Area.
C
D
E
F
L
R
18
20
25
30
35
40
45
50
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
2
3
4
5
6
7
8
9
10

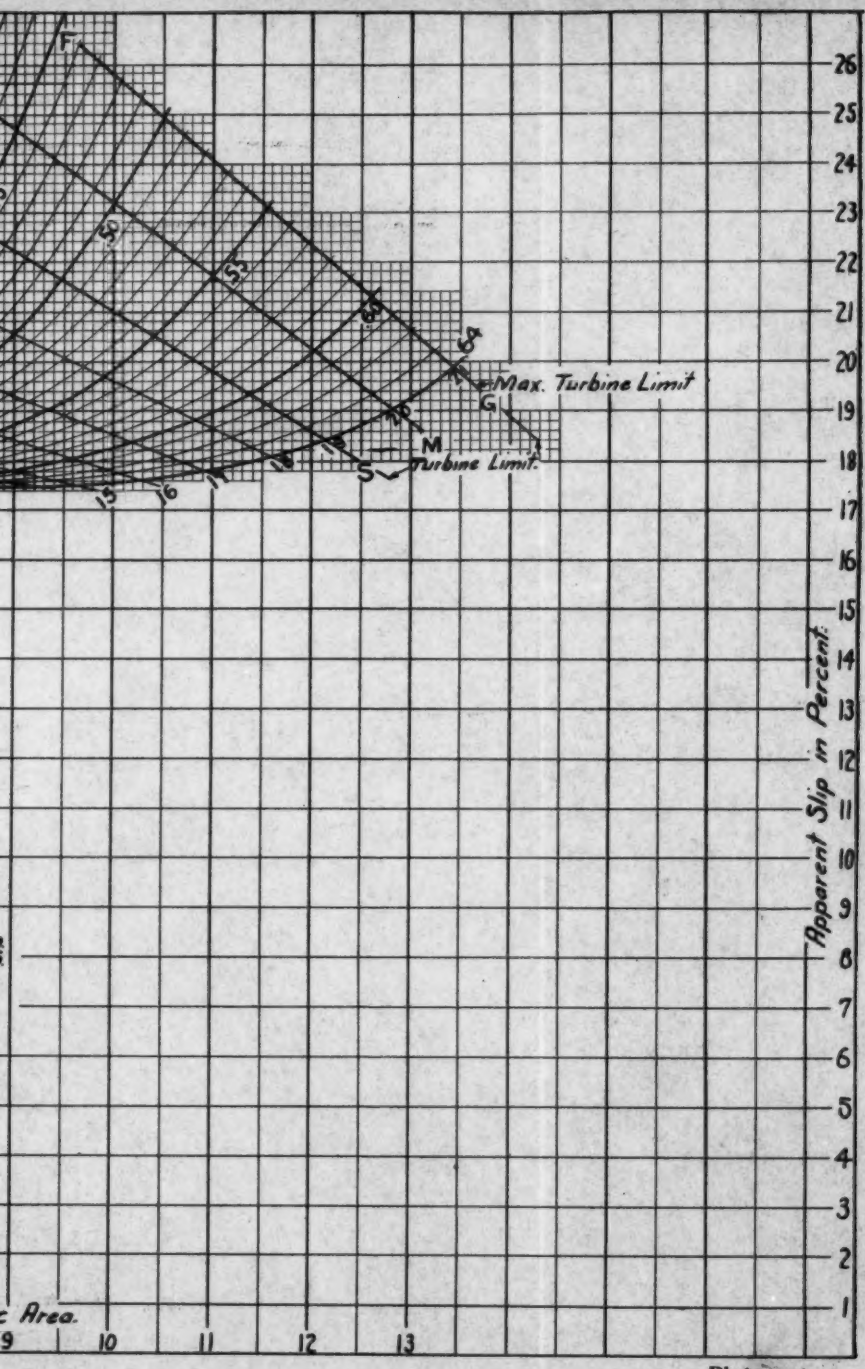
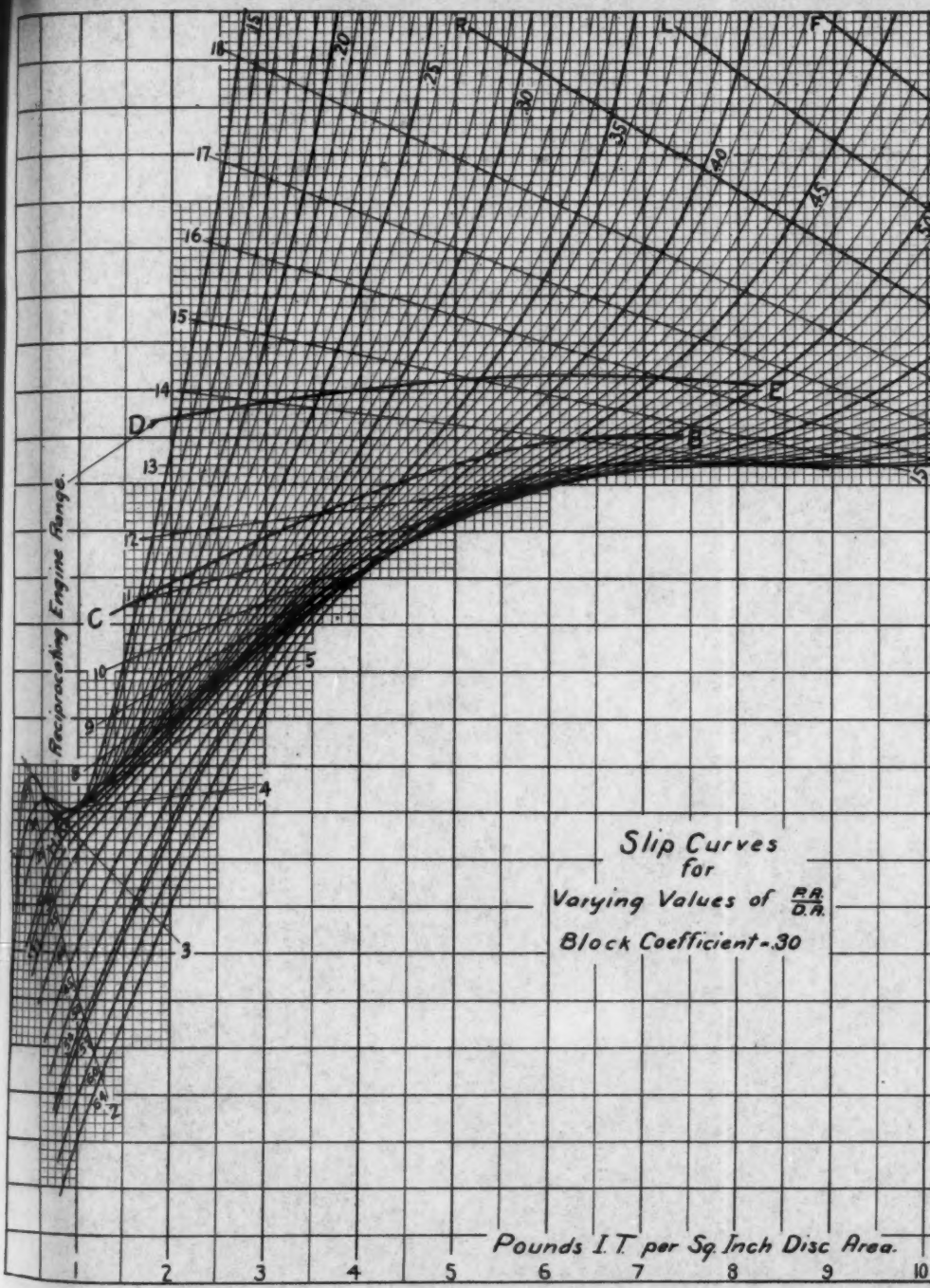

F
50
55
60
64
G
Max. Turbine Limit
M
Turbine Limit.
S
15
16
17
18
Apparent Slip in Percent.
26
25
24
23
22
21
20
19
18
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
Area.
9
10
11
12
13

Plate 7.


Reciprocating Engine Range.
A
B
C
D
E
F
Slip Curves
for
Varying Values of P.R. / D.R.
Block Coefficient - .30
Pounds I.T. per Sq. Inch Disc Area.
1
2
3
4
5
6
7
8
9
10

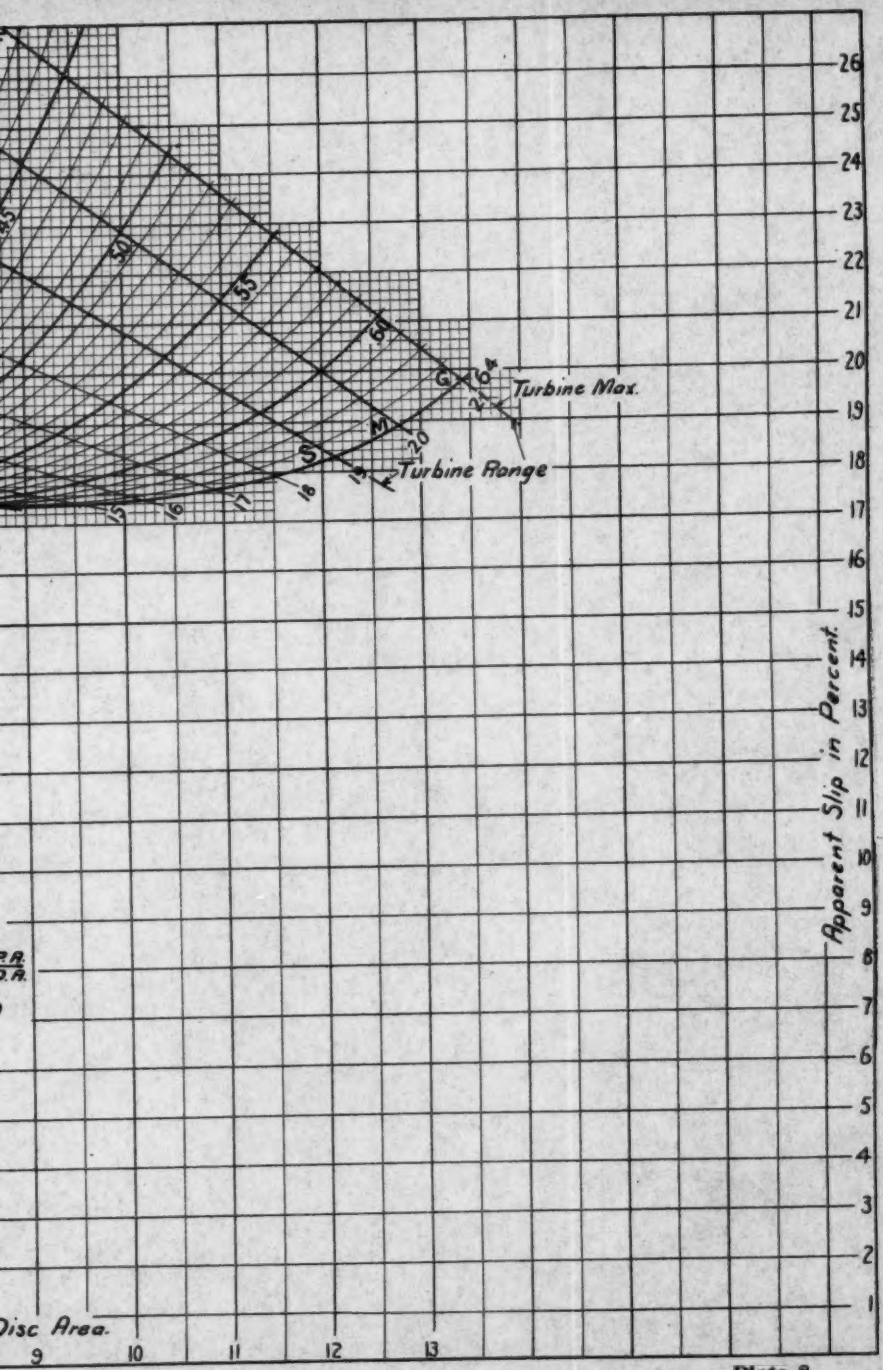
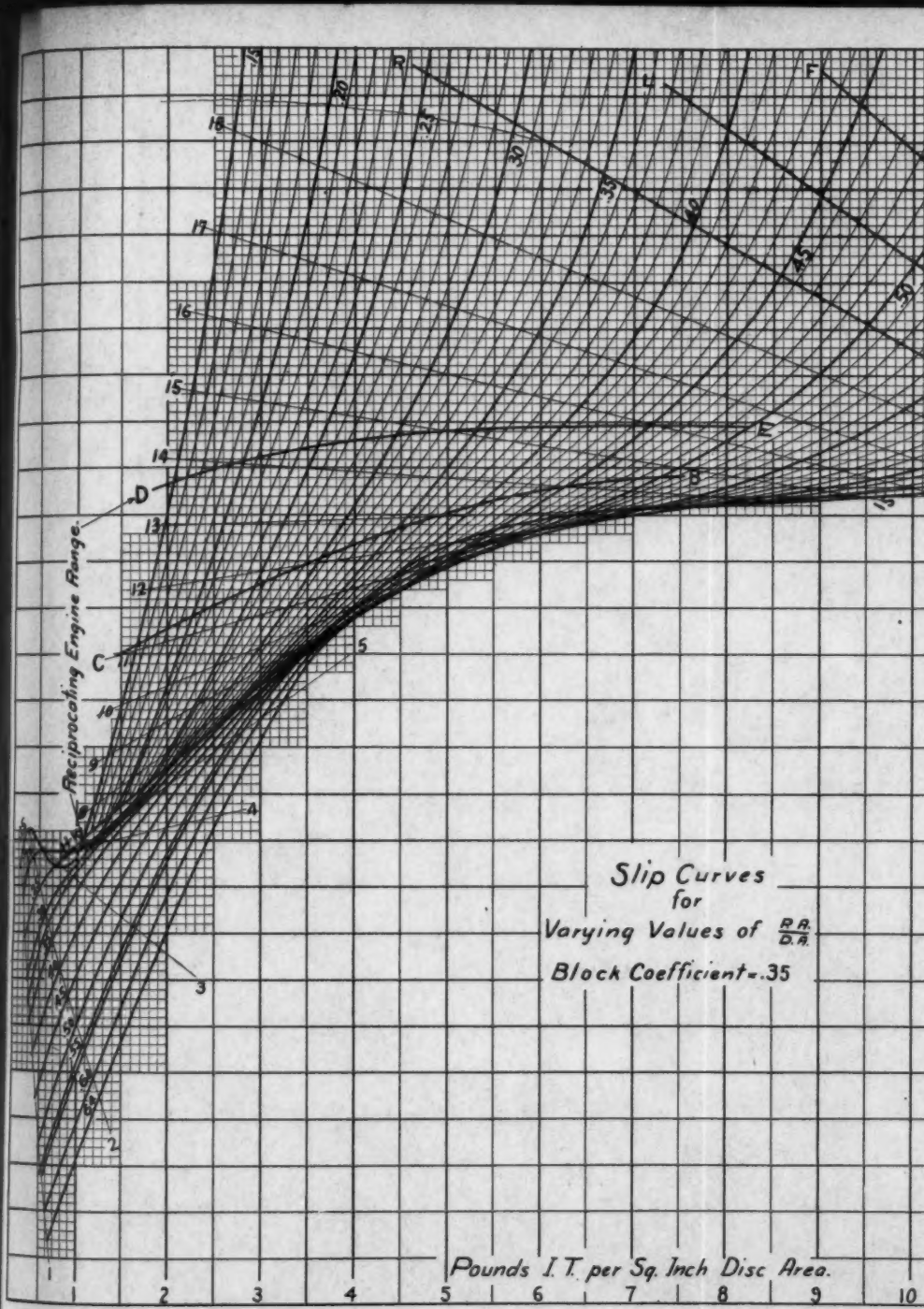

Turbine Max.
Turbine Range
Apparent Slip in Percent.
Disc Area.
9
10
11
12
13

Plate 8.


Reciprocating Engine Range.
Slip Curves
for
Varying Values of $\frac{P.A.}{D.A.}$
Block Coefficient = .35
Pounds I.T. per Sq. Inch Disc Area.
A
B
C
D
E
F
R
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20

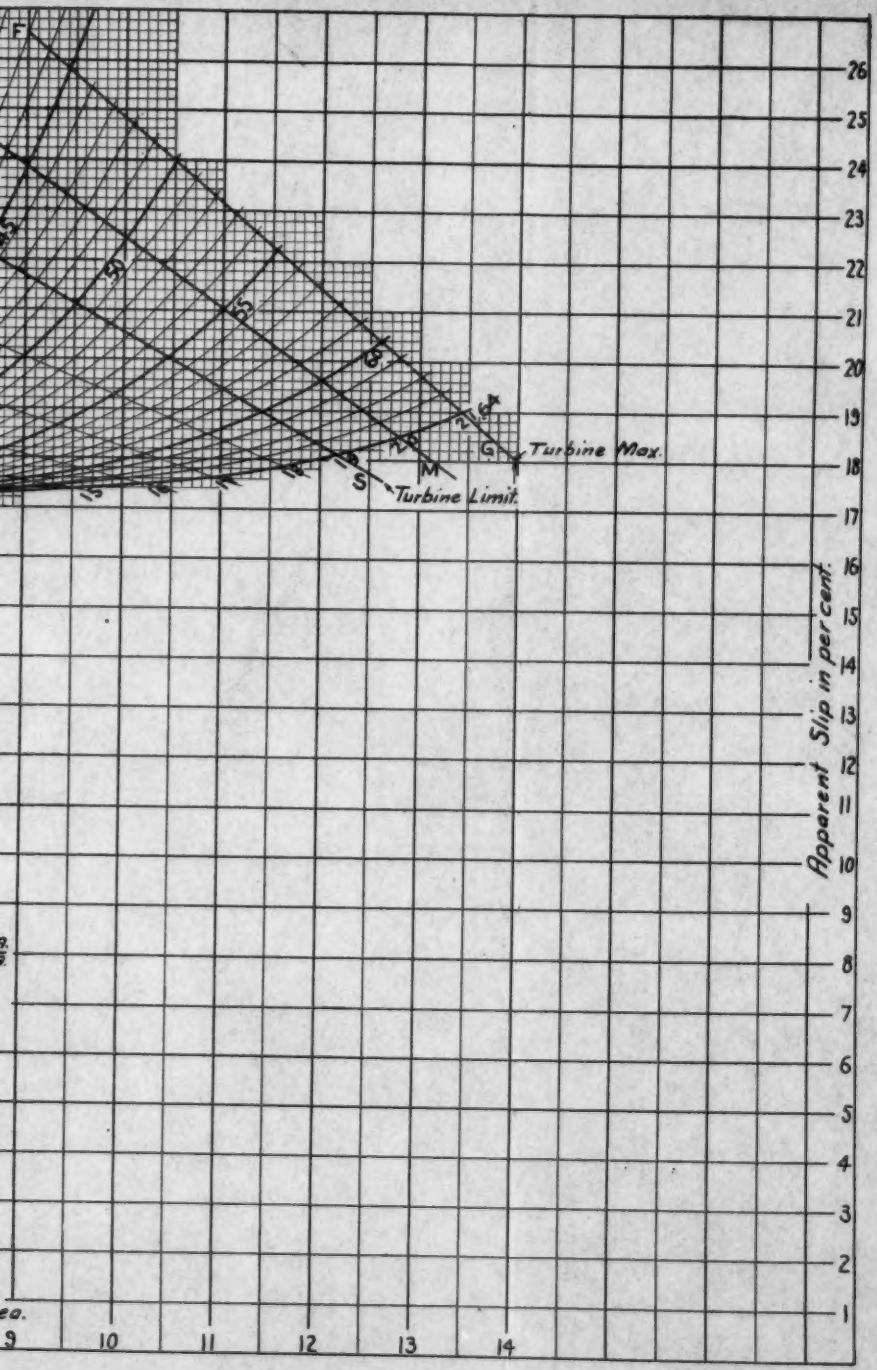
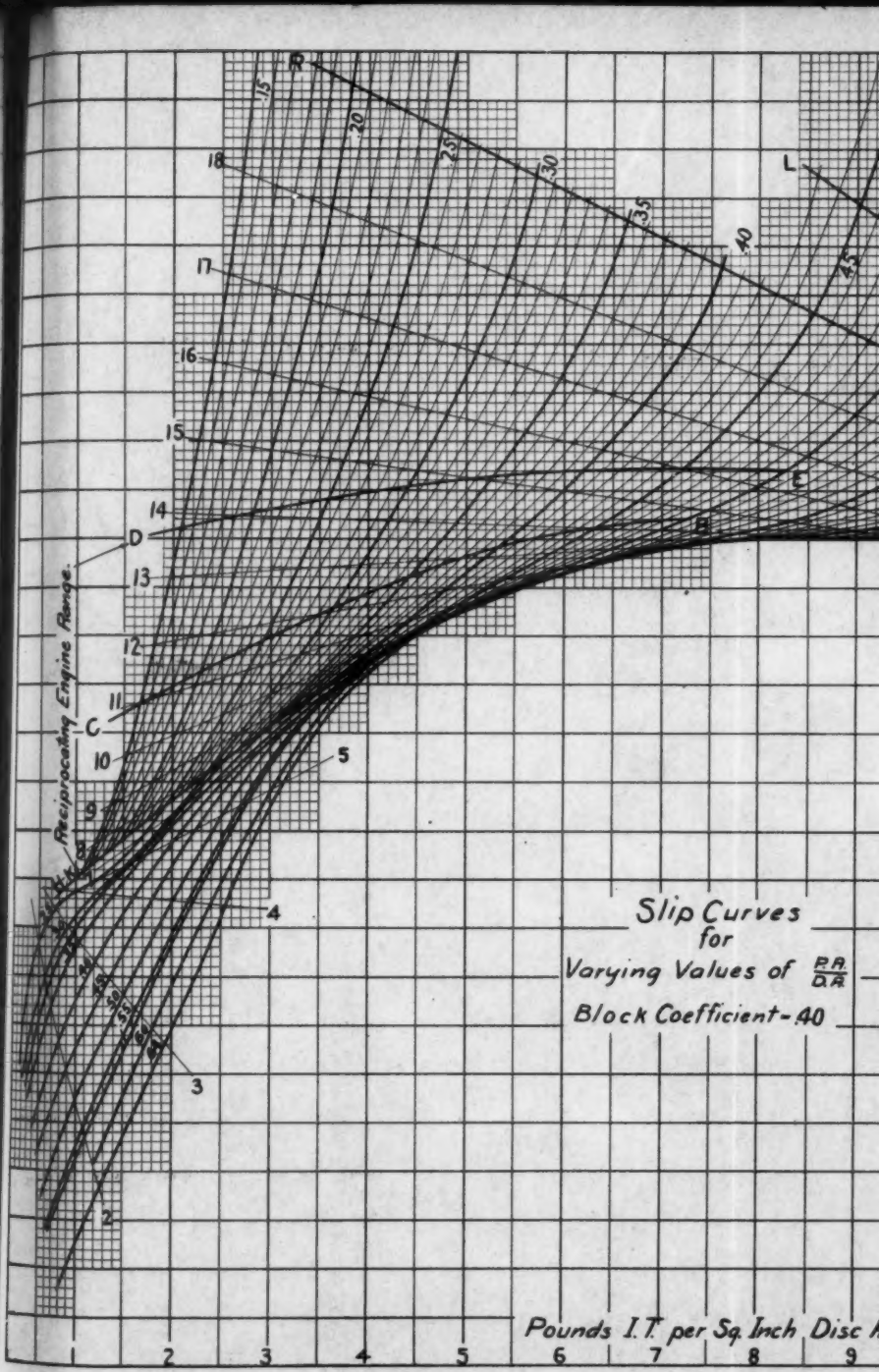

F
G
M
S
Turbine Max.
Turbine Limit.
Apparent Slip in per cent.
26
25
24
23
22
21
20
19
18
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
9
10
11
12
13
14

Plate 9.


R
L
.15
.20
.25
.30
.35
.40
.45
18
17
16
15
14
13
12
11
10
9
8
D
C
B
E
5
4
3
2
Reciprocating Engine Range.
Slip Curves
for
Varying Values of P.A / Q.R
Block Coefficient - .40
Pounds I.T. per Sq. Inch Disc
1
2
3
4
5
6
7
8
9

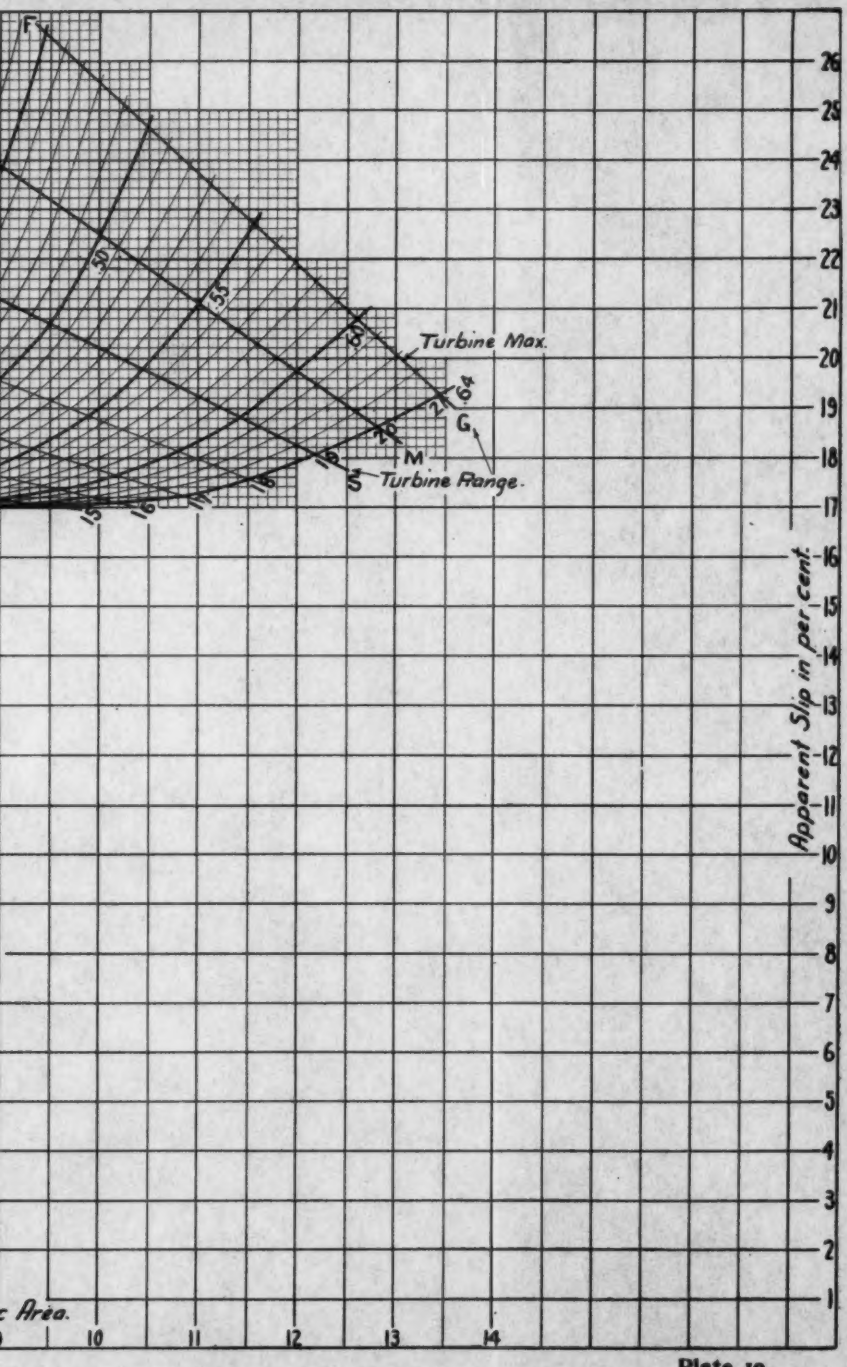
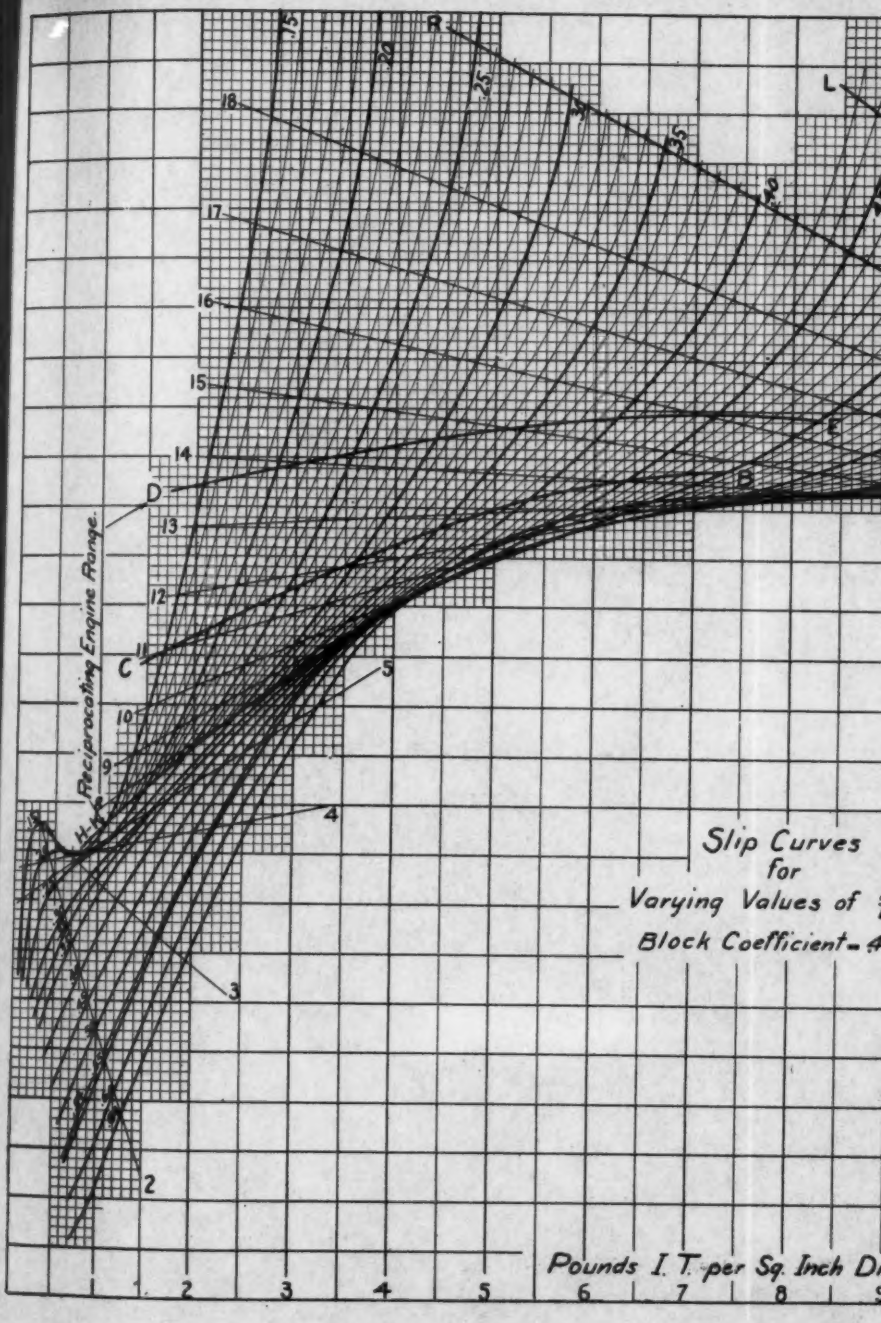

F
Turbine Max.
G
M
Turbine Range.
Apparent Slip in per cent.
26
25
24
23
22
21
20
19
18
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
10
11
12
13
14

Plate 10.


Reciprocating Engine Range
Slip Curves
for
Varying Values of
Block Coefficient - 4
Pounds I. T. per Sq. Inch D
R
L
D
C
B
E
18
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
2
3
4
5
6
7
8

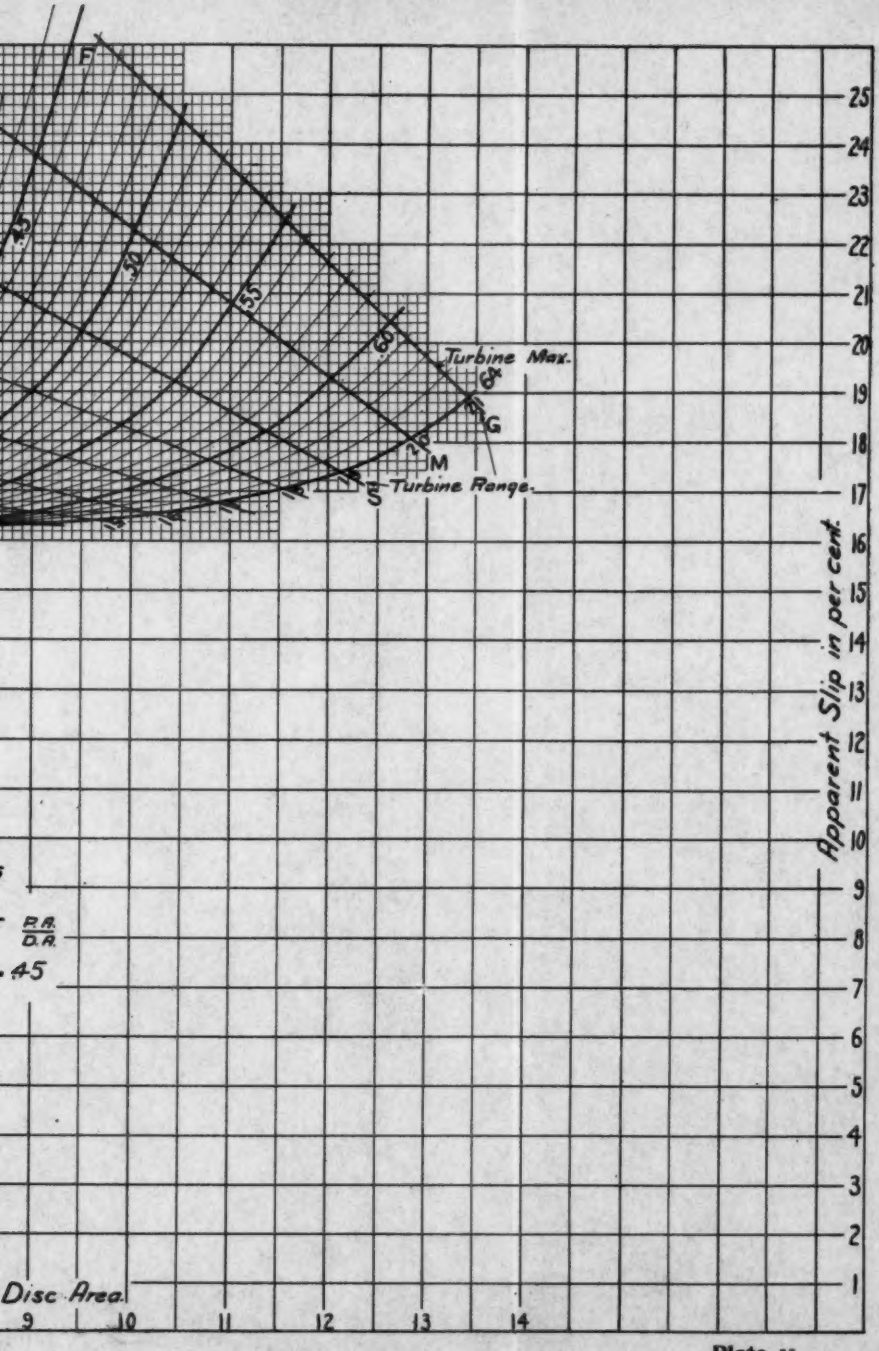
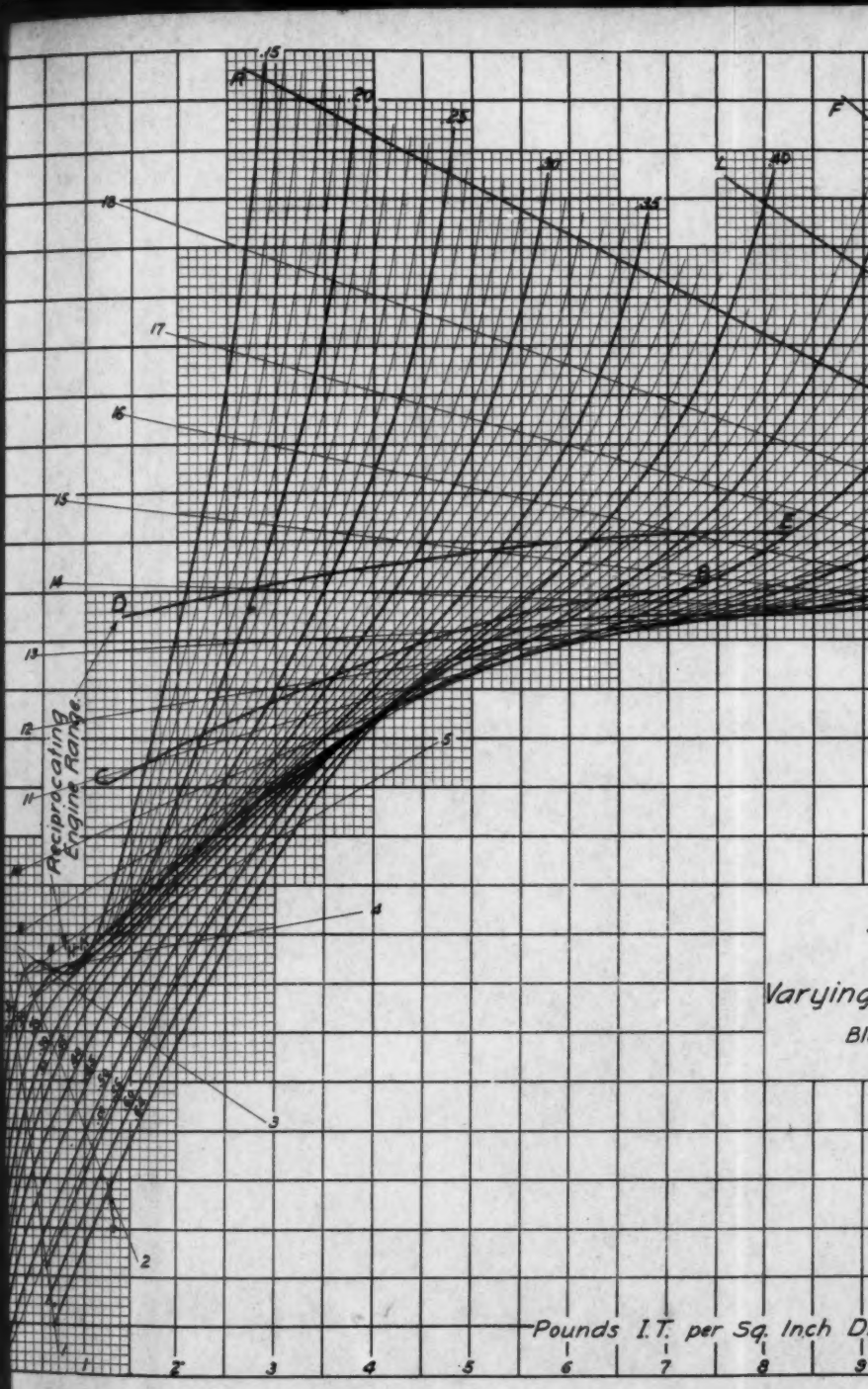
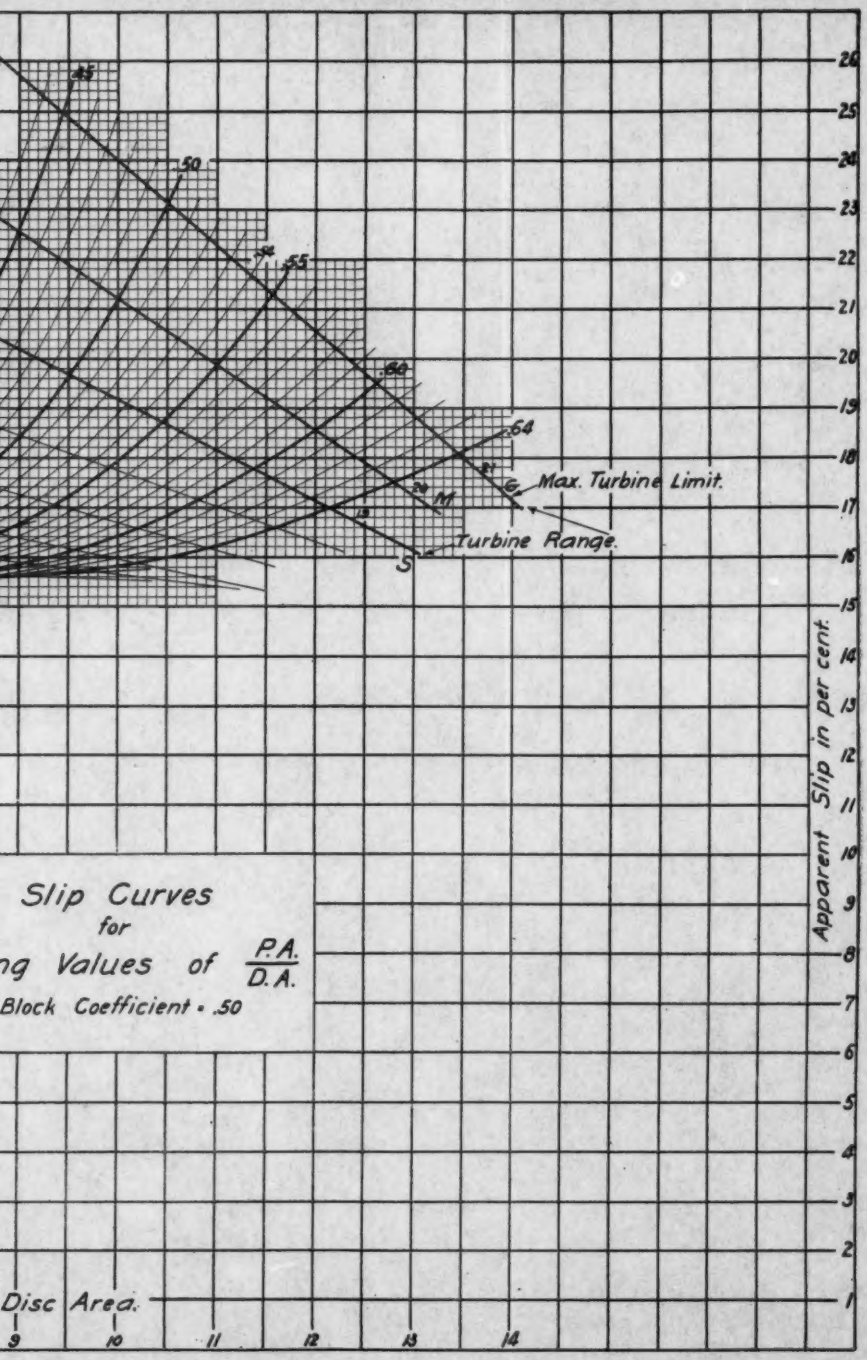

F
30
55
60
Turbine Max.
G
M
Turbine Range.
Apparent Slip in per cent.
25
24
23
22
21
20
19
18
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
P.A.
D.A.
.45
Disc Area
9
10
11
12
13
14

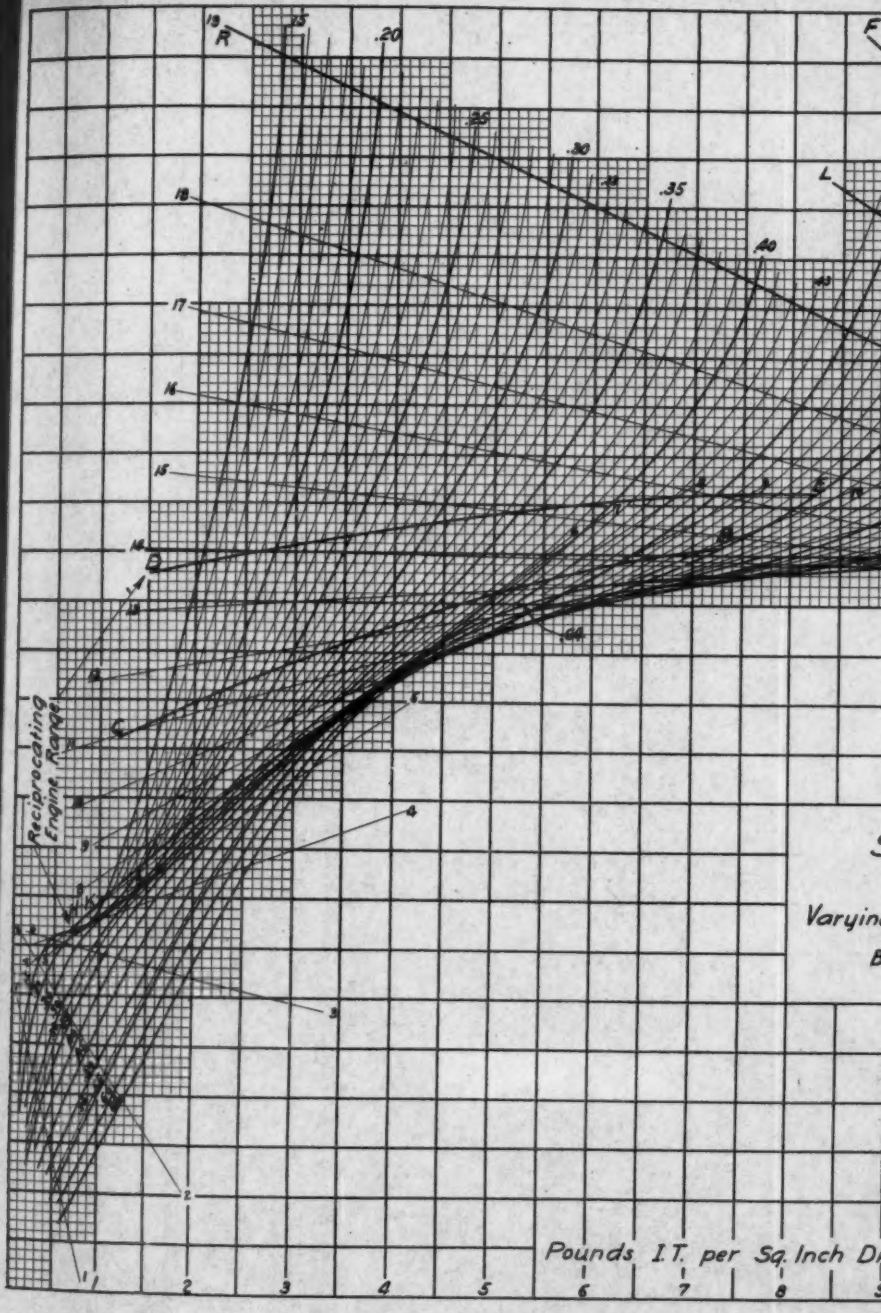
Plate II.


Reciprocating Engine Range
Varying
Pounds I.T. per Sq. Inch D
A
B
C
D
E
F
1
2
3
4
5
15
20
25
30
35
40
18
17
16
15
14
13
12
11
1
2
3
4
5
6
7
8
9



Slip Curves
for
ng Values of $\frac{P.A.}{D.A.}$
Block Coefficient = .50

Plate 12.


Reciprocating Engine Range
Pounds I.T. per Sq. Inch
1
2
3
4
5
6
7
8

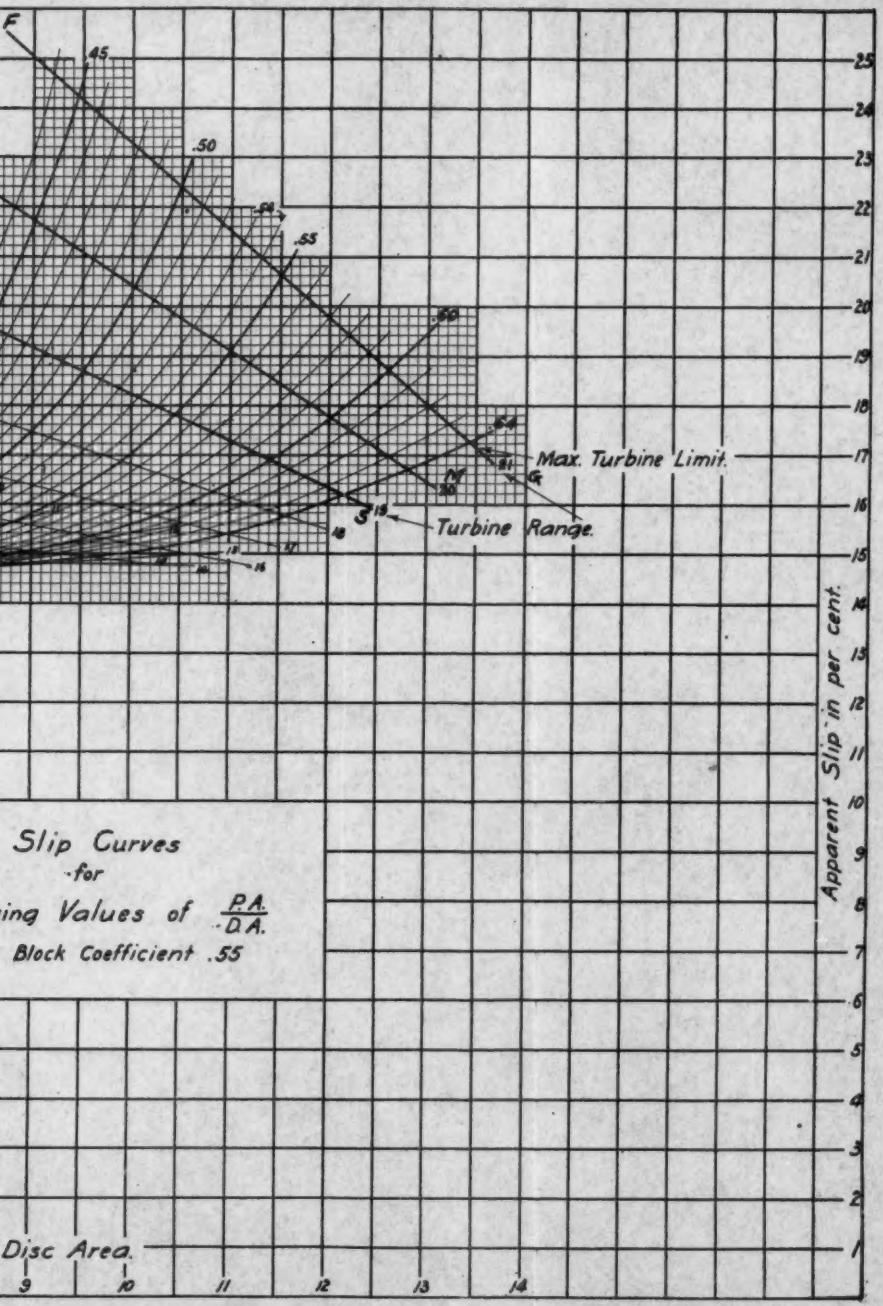
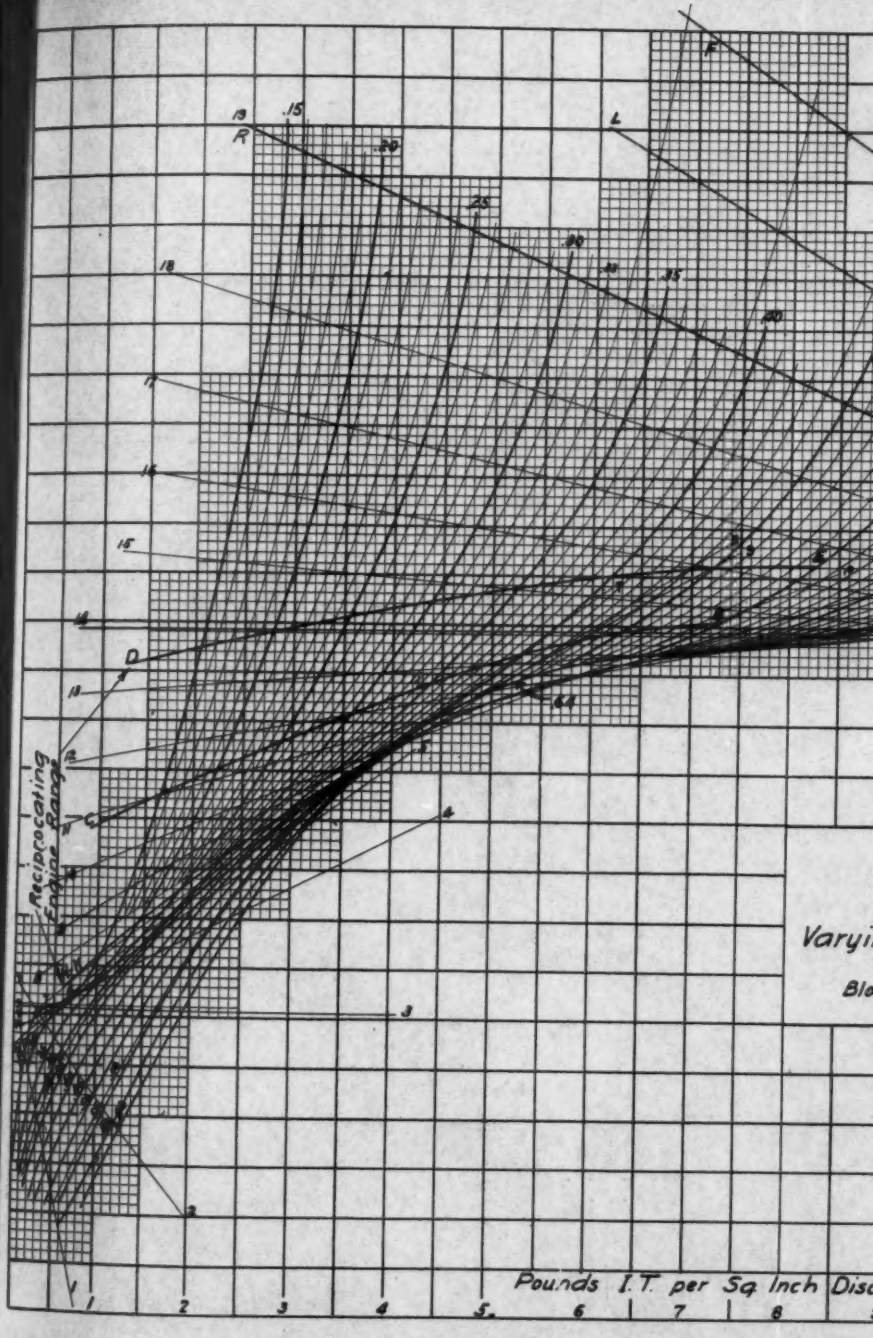

F
.45
.50
.54
.55
.60
.64
Max. Turbine Limit.
Turbine Range.
Slip Curves
for
ing Values of P.A. / D.A.
Block Coefficient .55
Apparent Slip in per cent.
25
24
23
22
21
20
19
18
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
Disc Area.
9
10
11
12
13
14

Plate 13.


Reciprocating Engine Range
Pounds I.T. per Sq Inch
1
2
3
4
5
6
7
8
R
L
D
C

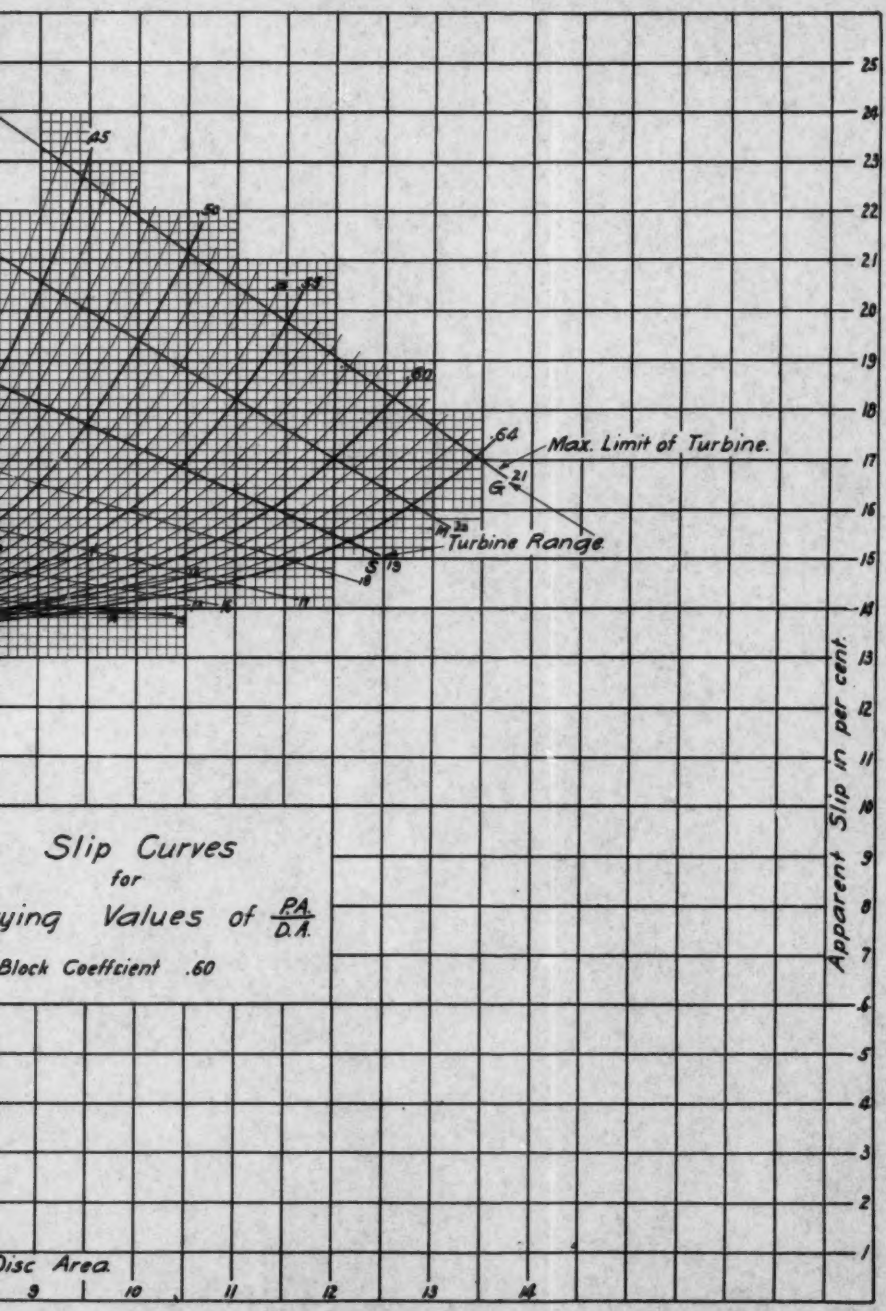
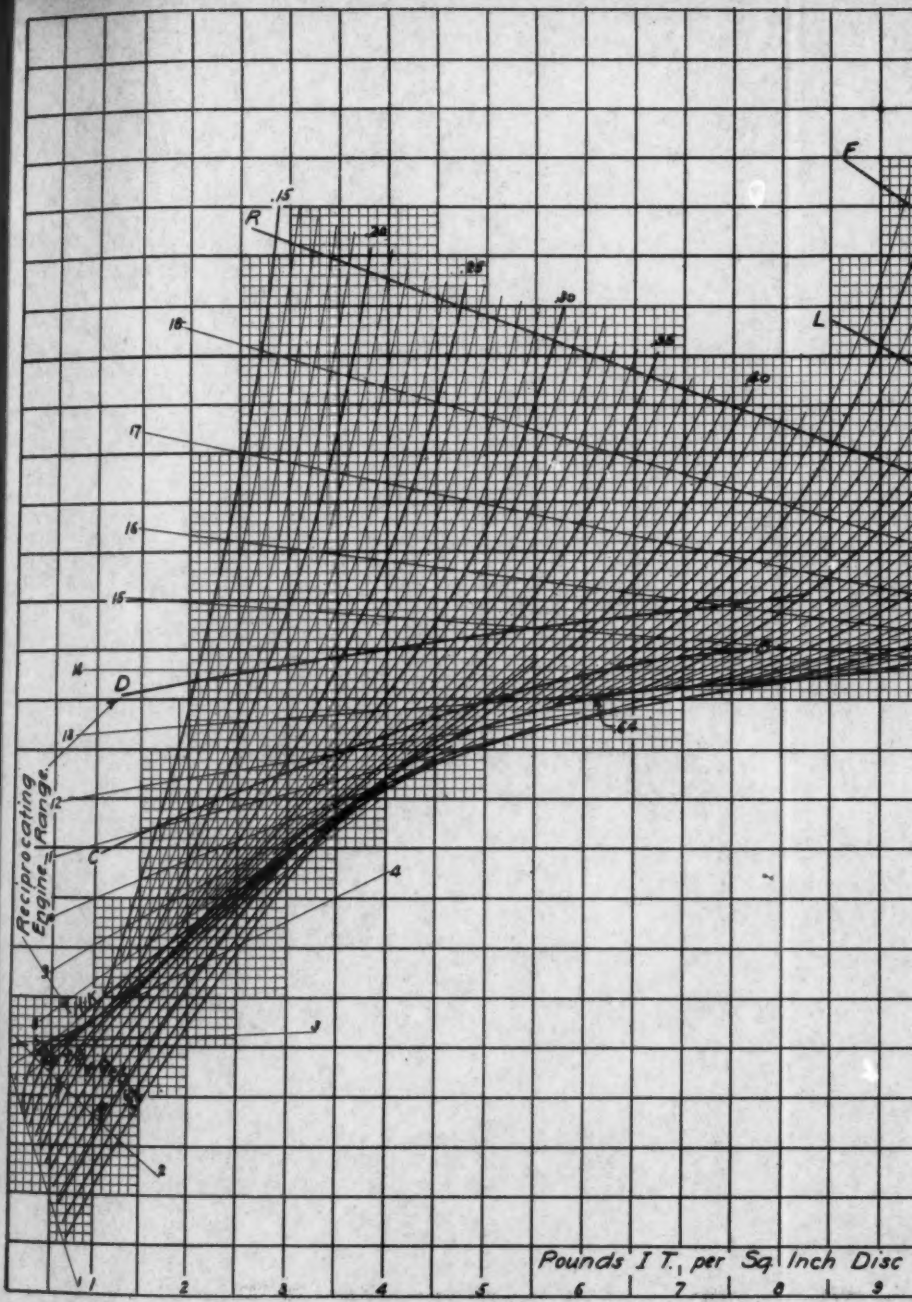

Max. Limit of Turbine.
Turbine Range
Slip Curves
for
ying Values of PA/D.A.
Block Coefficient .60
Apparent Slip in per cent.
Disc Area
9
10
11
12
13
14

Plate 14.


E
R
L
D
C
Reciprocating Engine Range
Pounds I T. per Sq. Inch Disc
1
2
3
4
5
6
7
8
9

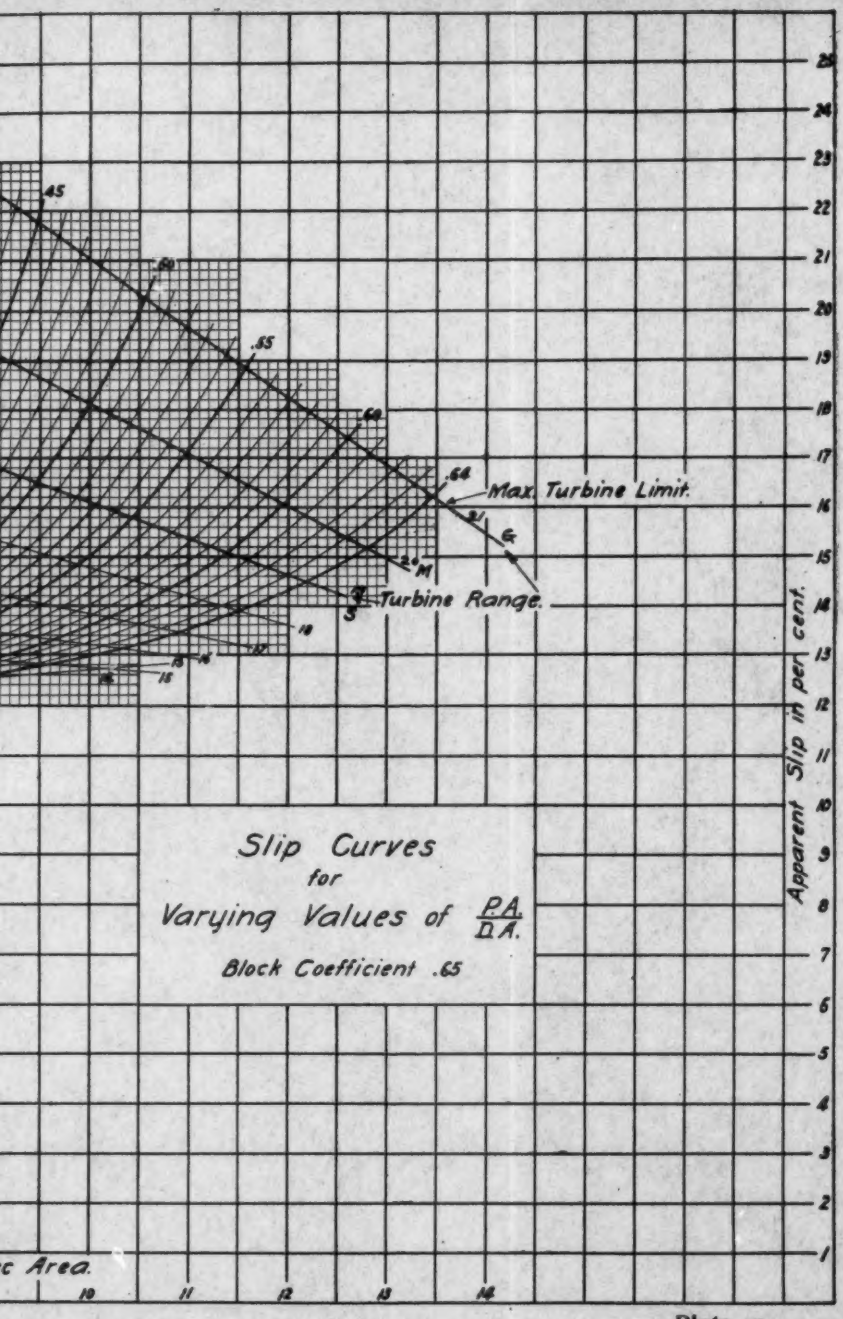
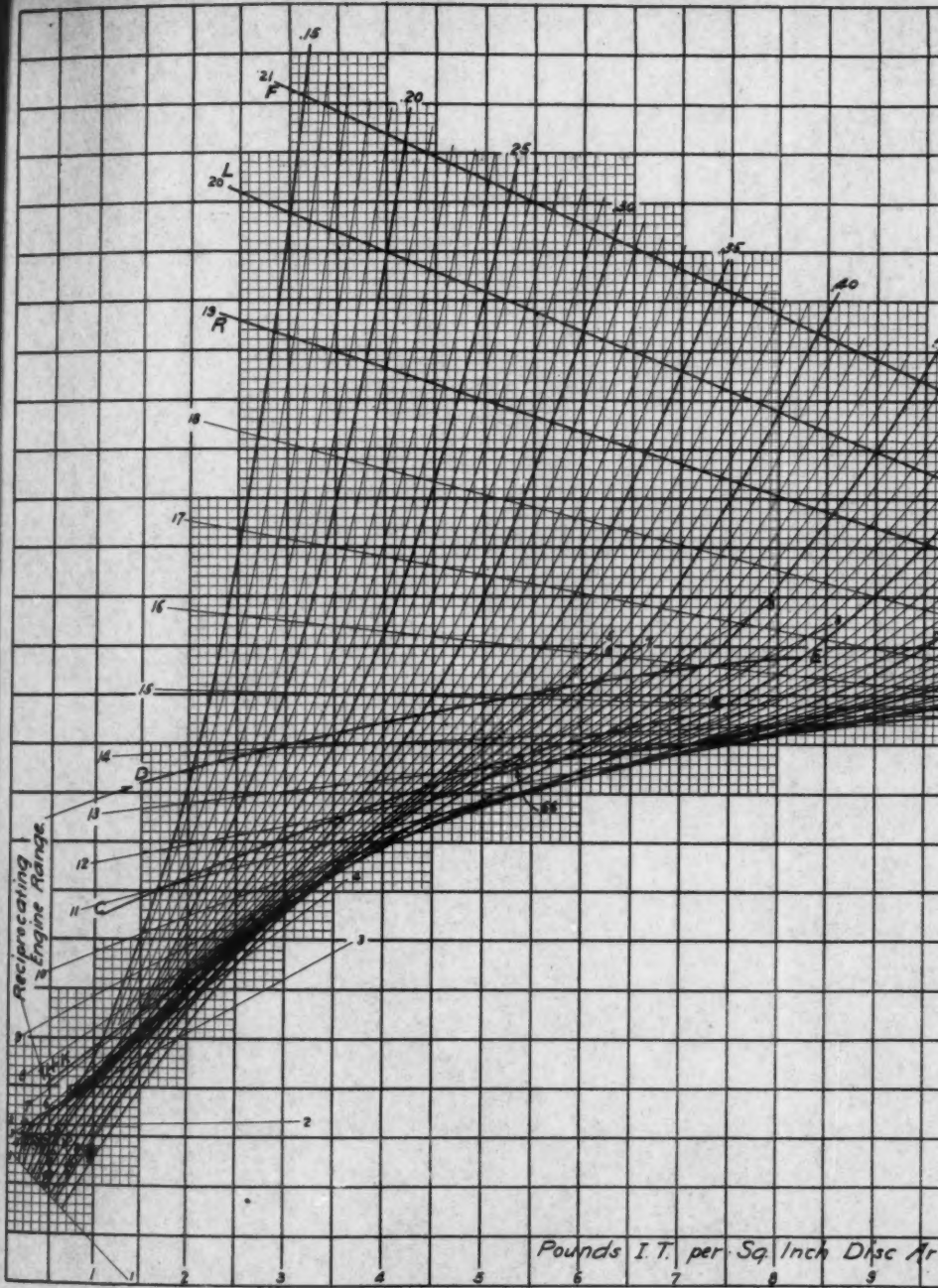

.45
.50
.55
.60
.64
Max. Turbine Limit.
Turbine Range.
Slip Curves
for
Varying Values of P.A./D.A.
Block Coefficient .65
Apparent Slip in per cent.
c Area.

Plate 15.


Reciprocating Engine Range
Pounds I.T. per Sq. Inch Disc Ar

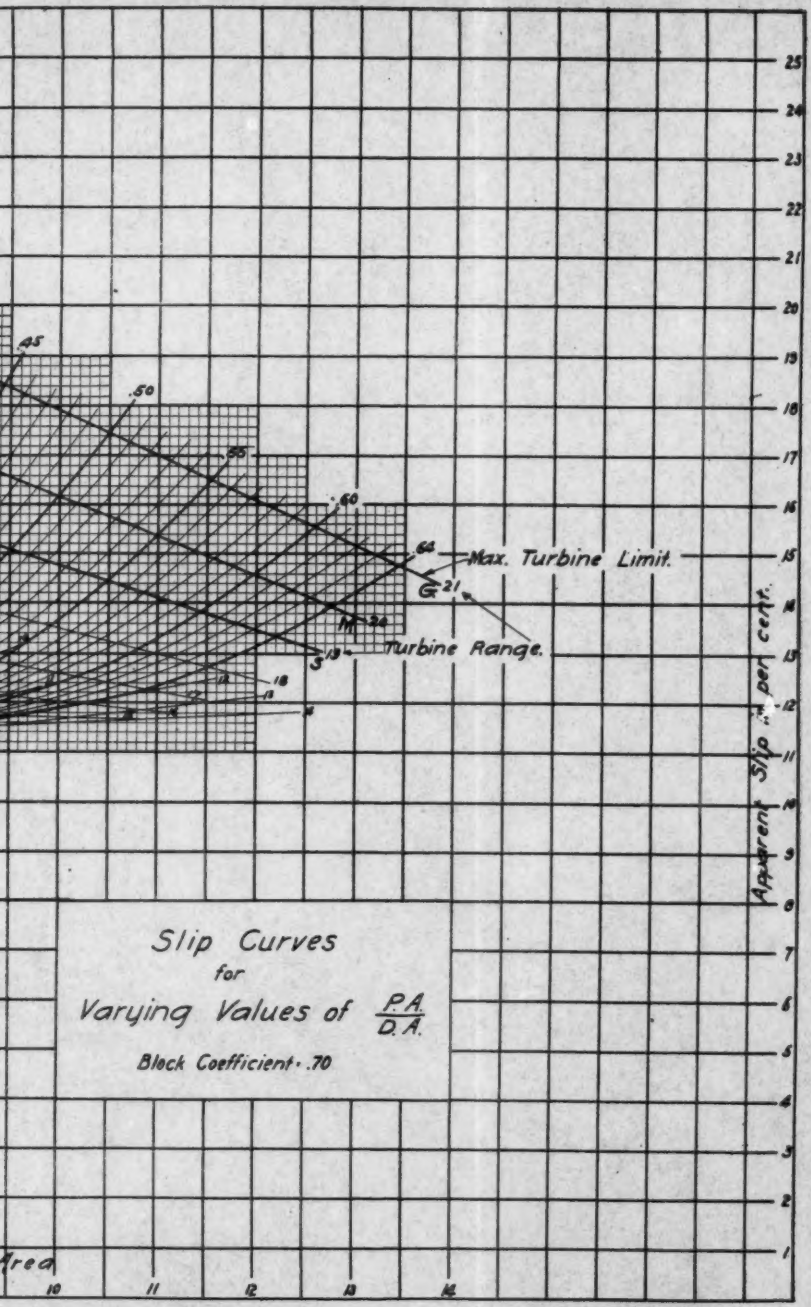
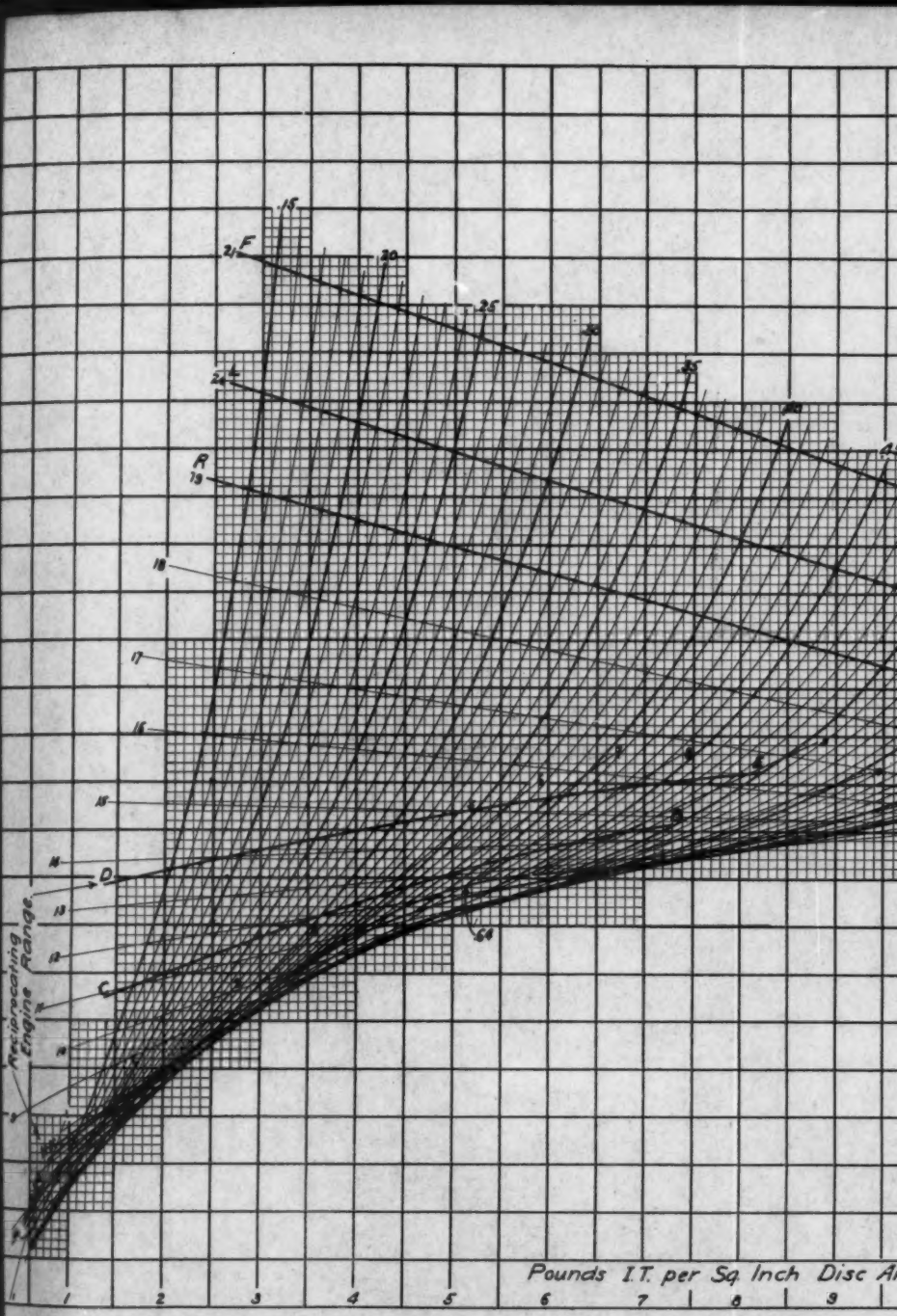

.45
.50
.55
.60
.65
Max. Turbine Limit.
.21
.24
Turbine Range.
.18
.15
Slip Curves
for
Varying Values of P.A. / D.A.
Block Coefficient - .70
Apparent Slip in per cent.
Area
10
11
12
13
14

Plate 16.


Reciprocating Engine Range
Pounds I.T. per Sq. Inch Disc Ar
1
2
3
4
5
6
7
8
9
D
C
E
R
15
20
25
30
35
40
45
50
55
60

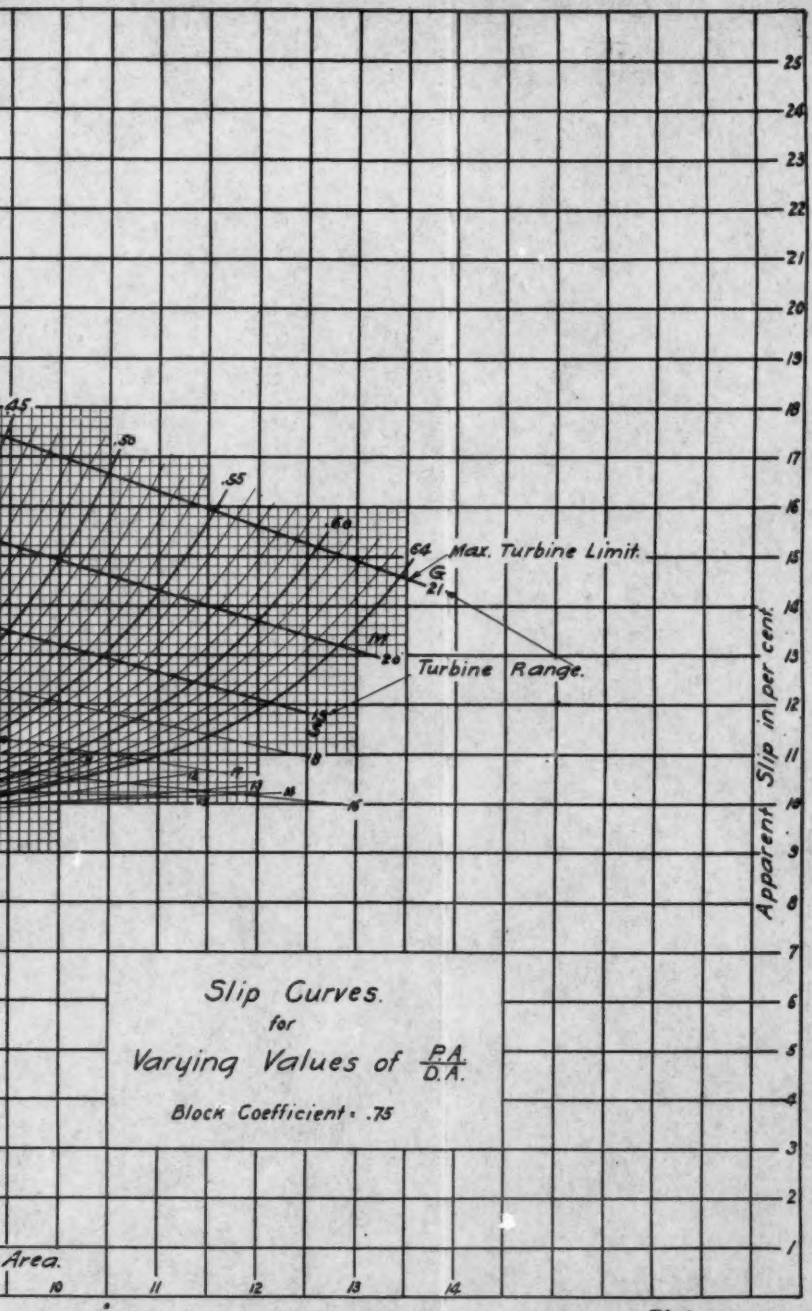
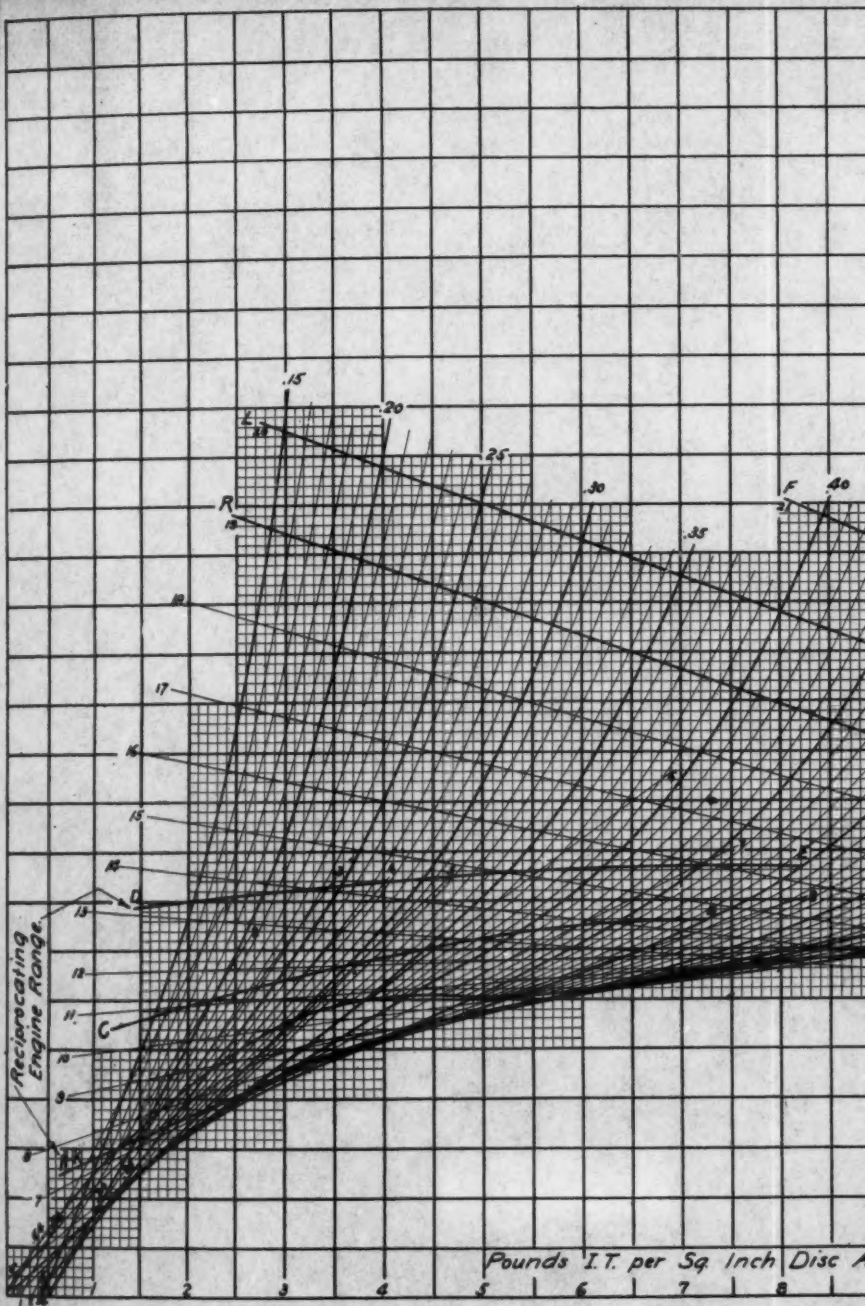

Slip Curves.
for
Varying Values of P.A./D.A.
Block Coefficient = .75
Max. Turbine Limit.
Turbine Range.
Apparent Slip in per cent.
Area.
.45
.50
.55
.60
.64
G
21
M
20
10
11
12
13
14
15
16
17
18
19
20
21
22
23
24
25

Plate 17.


.15
.20
.25
.30
.35
.40
L
R
F
Reciprocating Engine Range.
Pounds I.T. per Sq. Inch
1
2
3
4
5
6
7
8

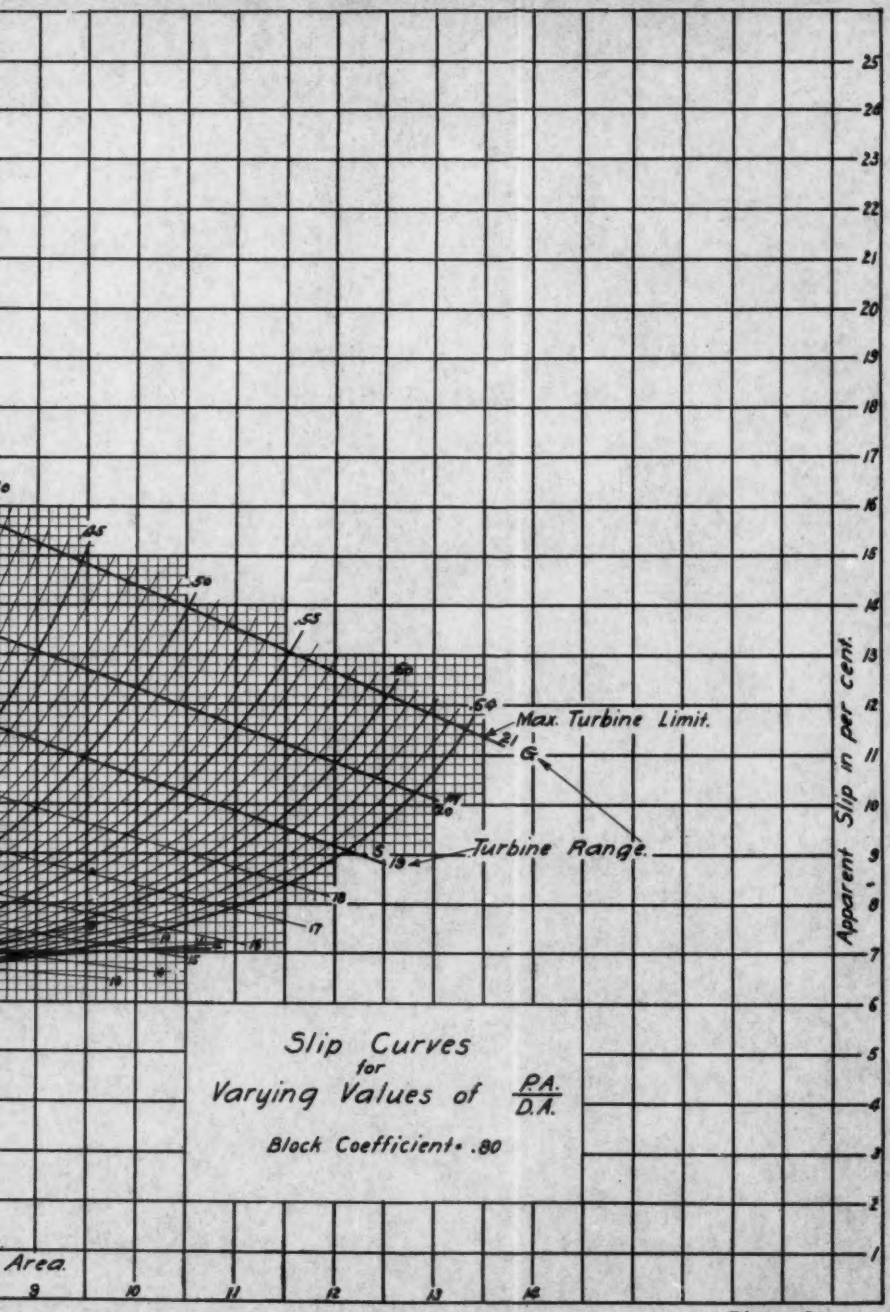
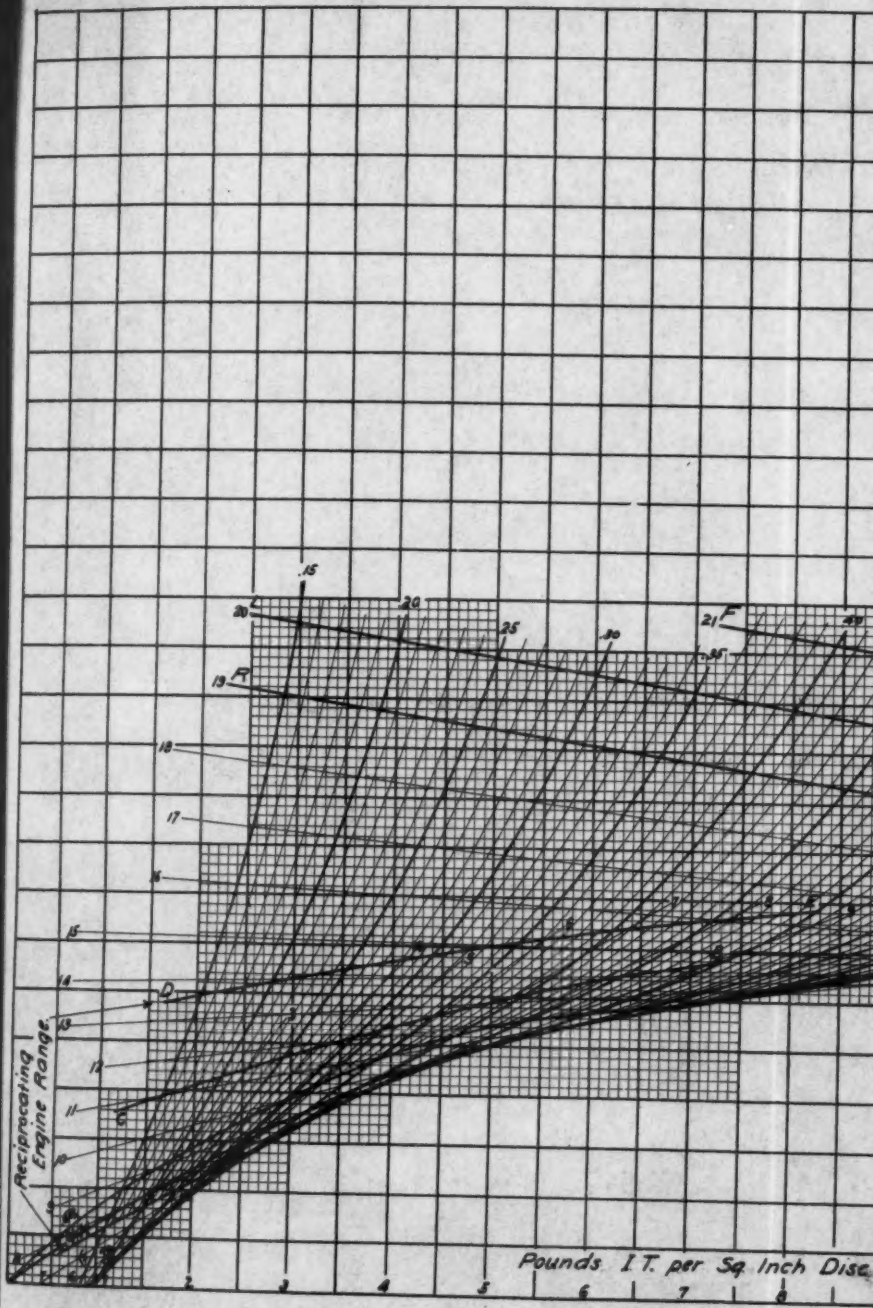

Apparent Slip in per cent.
Max. Turbine Limit.
Turbine Range.
Slip Curves
for
Varying Values of P.A./D.A.
Block Coefficient = .80
Area.

Plate 18.


Reciprocating Engine Range.
Pounds I.T. per Sq. Inch Disc.

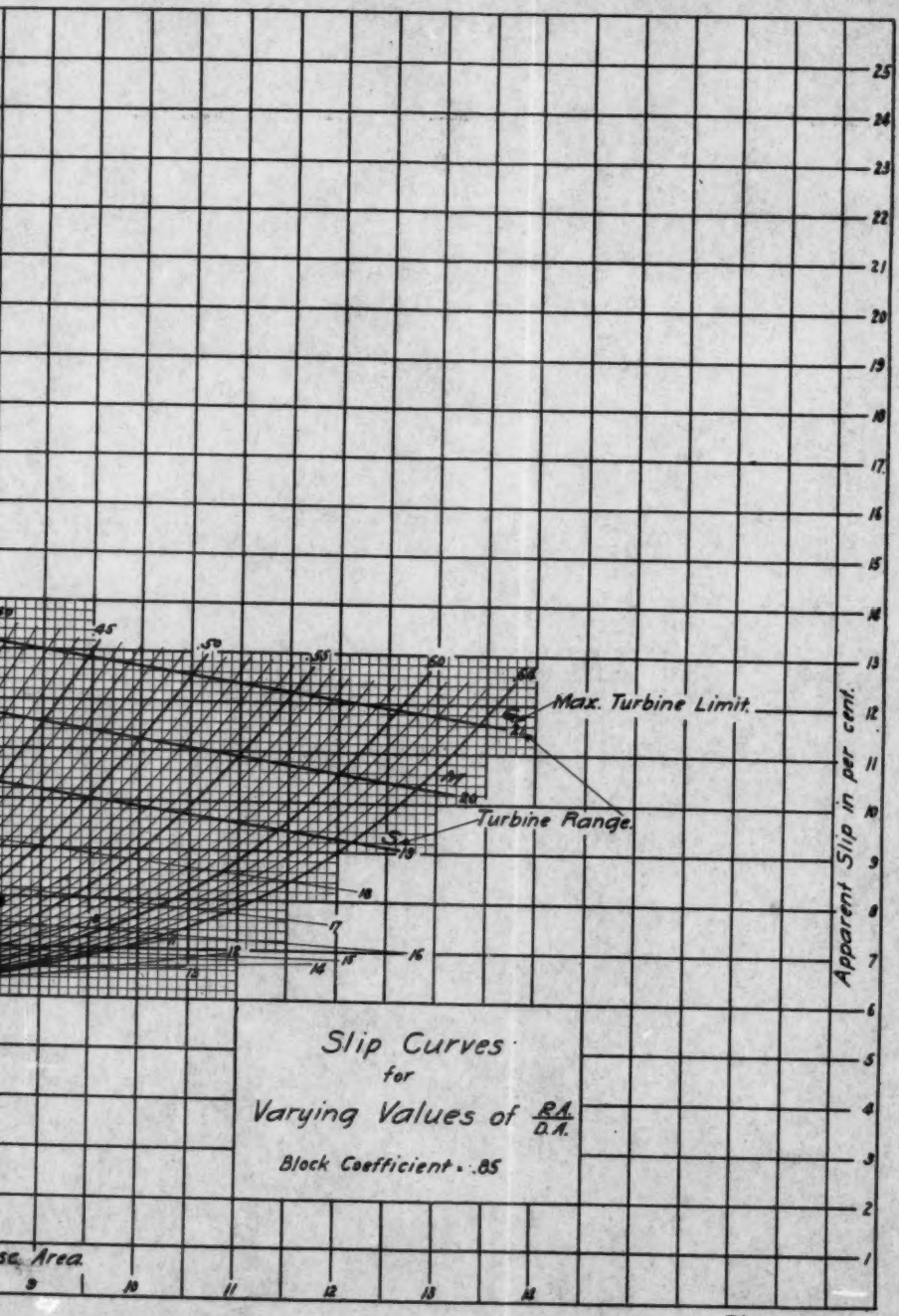
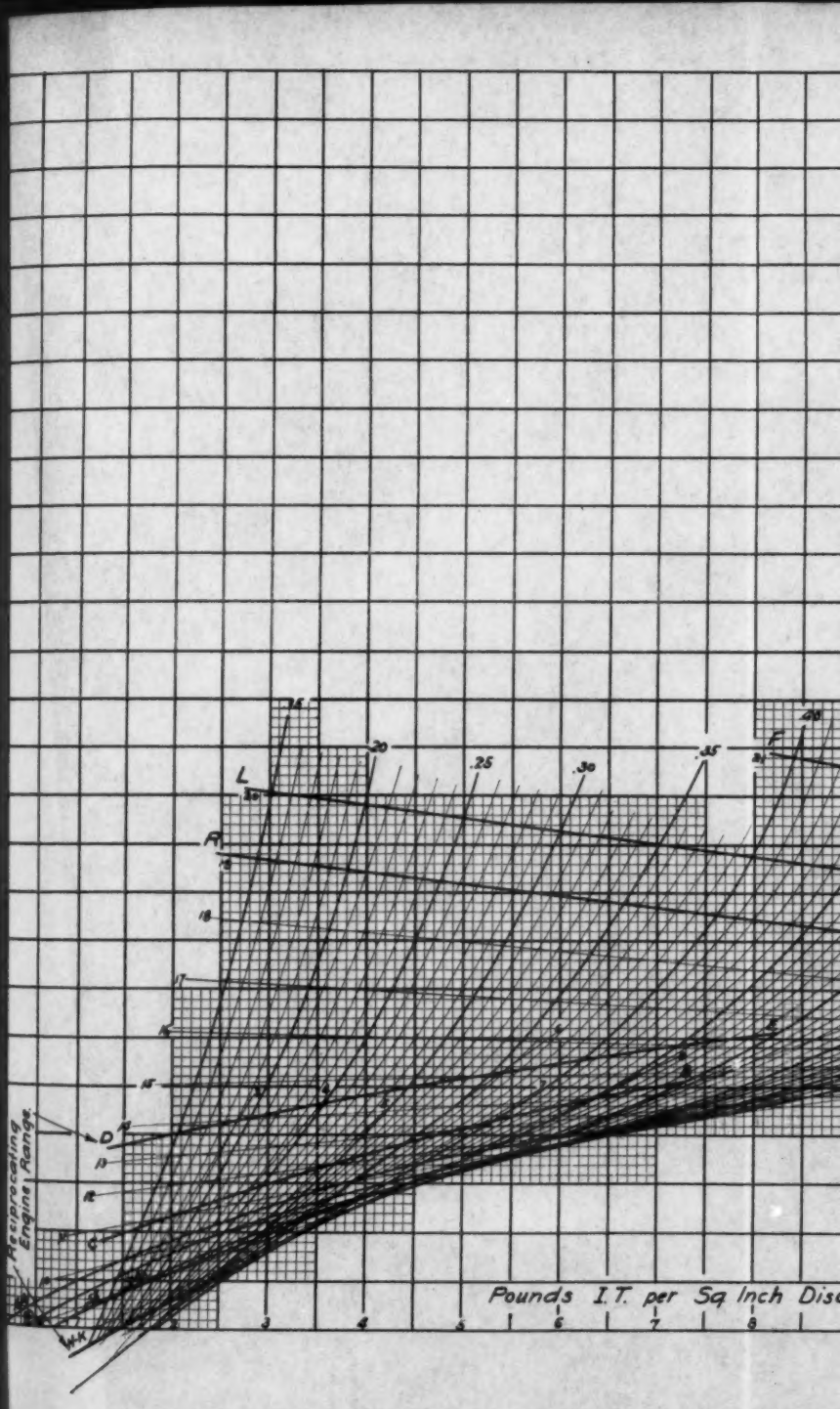

Max. Turbine Limit.
Turbine Range.
Apparent Slip in per cent.
Slip Curves
for
Varying Values of $\frac{R.A.}{D.A.}$
Block Coefficient = .85
sc. Area.

Plate 19.


Reciprocating Engine Range
Pounds I.T. per Sq Inch Disc
L
R
D
K
14
16
18
20
25
30
35
40
2
3
4
5
6
7
8

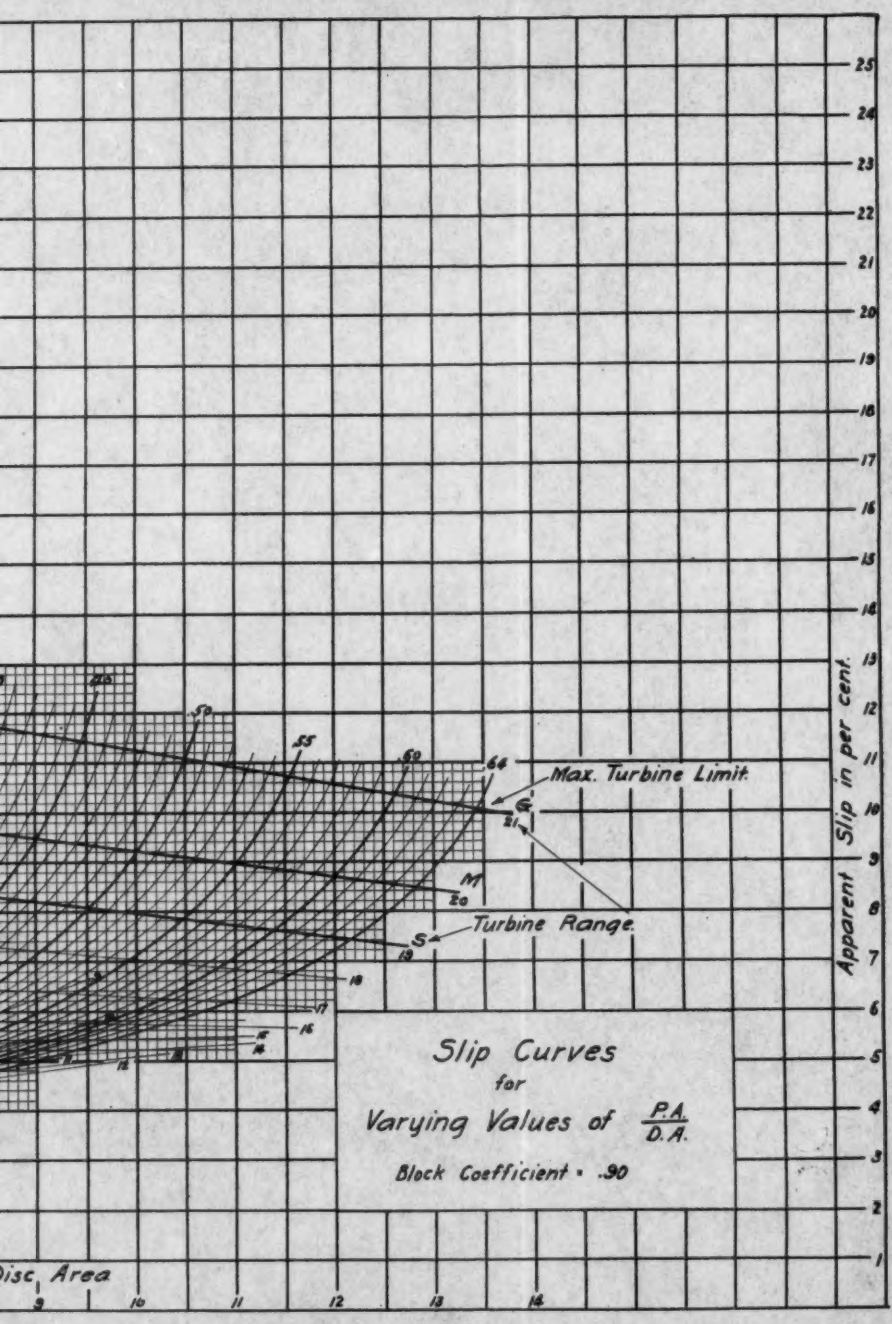
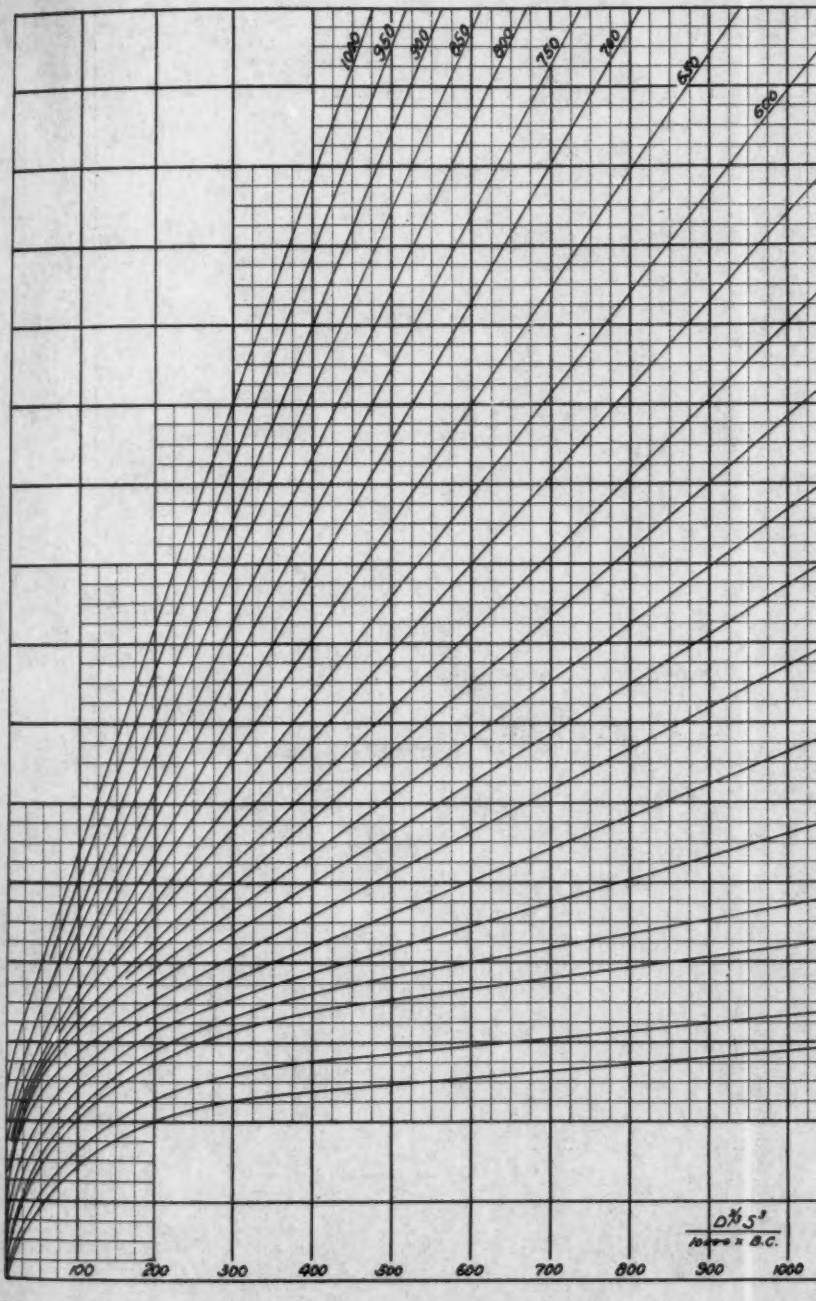
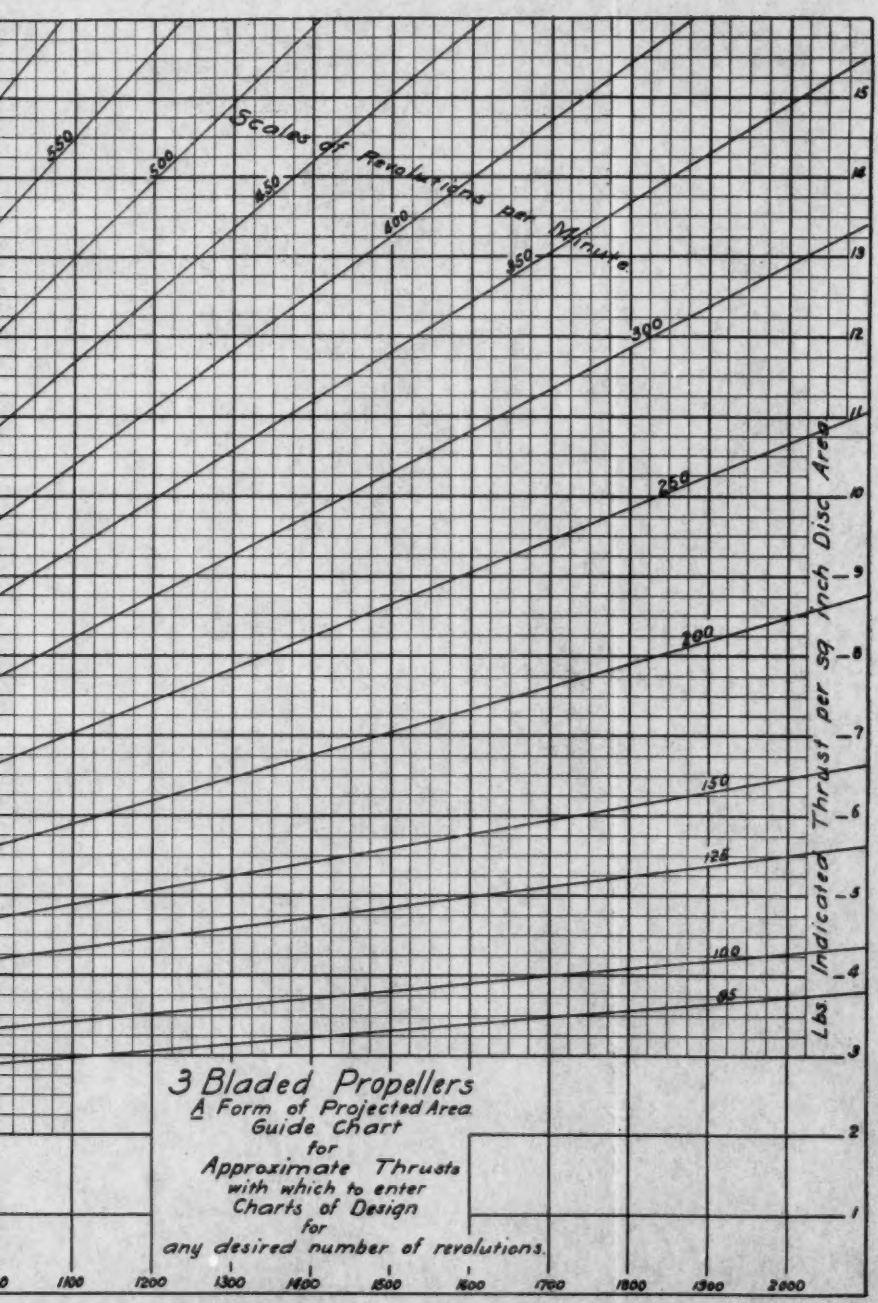

Slip Curves
for
Varying Values of P.A. / D.A.
Block Coefficient = .90
Max. Turbine Limit.
Turbine Range.
Apparent Slip in per cent.
Disc Area
45
50
55
60
64
G
21
M
20
S
19
18
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
9
10
11
12
13
14

Plate 20.


1000
950
900
850
800
750
700
650
600
$\frac{D^{\frac{1}{2}} S^{\frac{1}{2}}}{\text{Metric B.C.}}$
100
200
300
400
500
600
700
800
900
1000



3 Bladed Propellers

A Form of Projected Area Guide Chart for Approximate Thrusts with which to enter Charts of Design for any desired number of revolutions.

Plate 21.

PROPOSED STANDA

SCREW PROPELLERS

C.W.
COMMAND

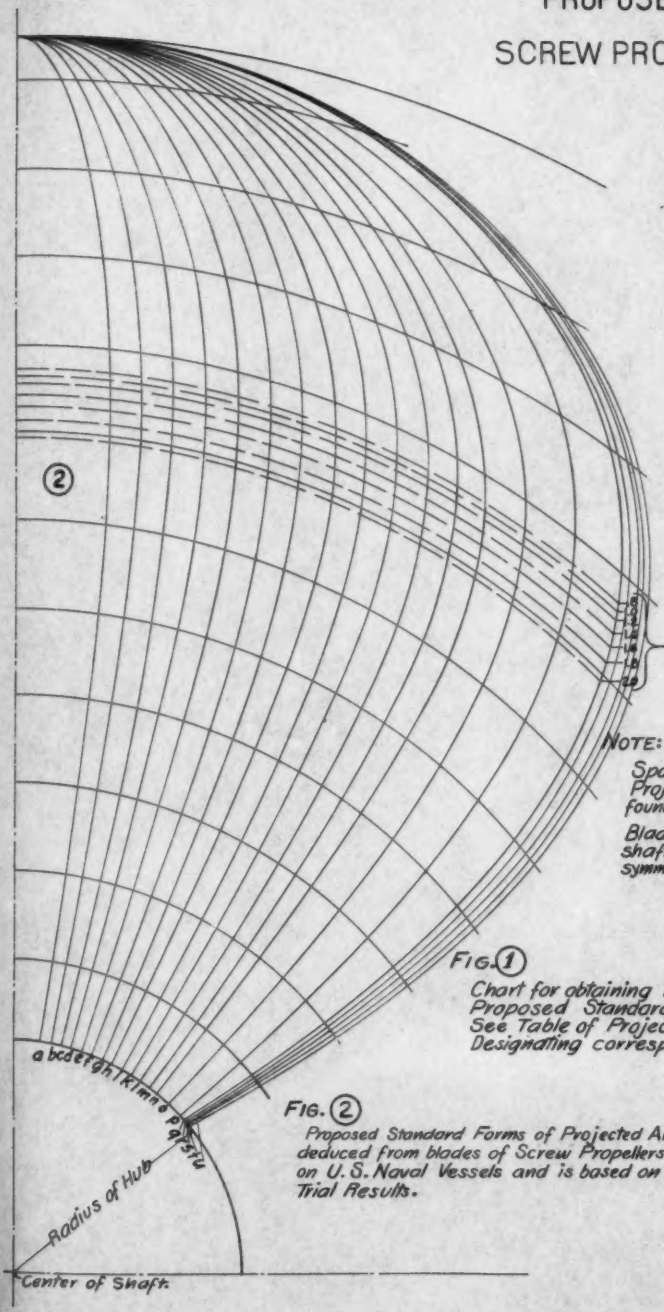


FIG. (1)
Chart for obtaining Ratios of Blade
Proposed Standard Form of Proje
See Table of Projected Area Ratios
Designating corresponding Form.

FIG. (2)
Proposed Standard Forms of Projected Areas deduced from blades of Screw Propellers used on U. S. Naval Vessels and is based on actual Trial Results.

...ARD FORM OF BLADES
-FOR-
...S OF U. S. NAVAL VESSELS.
BY
...V. DYSON
...NDER U.S.N.

...f greatest Elliptical Widths
...ng to Pitch Ratios.

... bounding lines a-b-c-etc. are
...rresponding to Pitch Ratios as
...n of Chart-Fig. I.
...r Vertical or Inclined to axis of
...e true pitch and be of
...ut Generatrix.

...des having
...jected Area.
...os for Letter
...

See Fig. 2-for Corresponding Form.	
Ratio- P.A. / D.A.	Designating Letters.
.097	a
.127	b
.159	c
.188	d
.221	e
.255	f
.283	g
.314	h
.346	i
.376	k
.408	l
.438	m
.467	n
.513	o
.565	p
.619	q
.64	r
.65	s
.66	t
.667	u

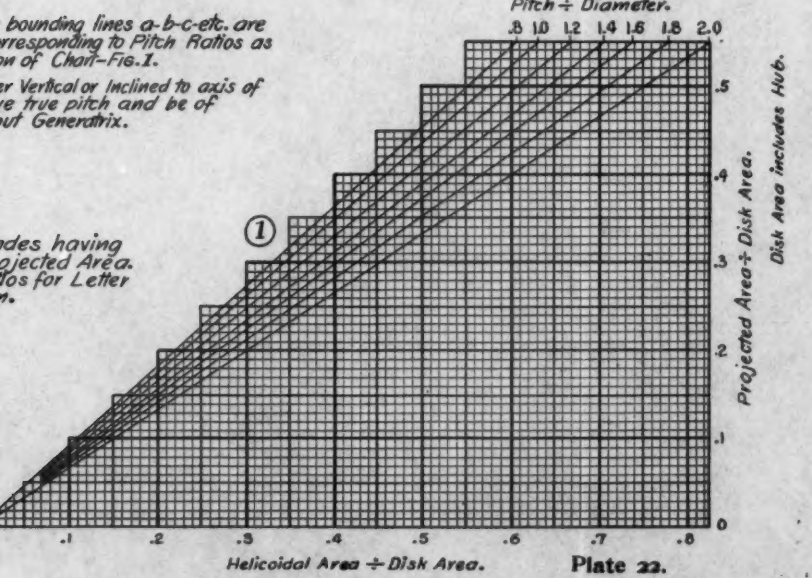


Plate 22.

A PLEA FOR A STANDARD ORGANIZATION OF THE ENGINEER DIVISION ABOARD SHIP, AND FOR A UNIFORM METHOD OF MANAGEMENT OF THE ENGINEER DEPARTMENT, WITH A SECTION DEVOTED TO THE APPLICATION OF SCIENTIFIC MANAGEMENT.

By Lieut. Comdr. W. B. Tardy, U. S. Navy, Member.

GENERAL DISCUSSION.

COMMERCIAL RIVALRY.

With the prevailing wage scale consequent upon the present high cost of living and the continuously increasuring prices of the new materials entering into manufactured products, industrial concerns are forced to practice rigid economies and constantly strive to increase their efficiency if they would maintain their prestige and continue to pay dividends to their stockholders.

The demands of the public coupled with the vigorous competition that exists among manufacturers of similar products tend to keep the quality high. Economies of production cannot be obtained by putting an inferior article on the market.

In the present day of almost instantaneous communication it is hardly conceivable that one concern can gain much advantage over another by placing orders for material at a better figure, or by selling the manufactured product at a higher price.

Nor can it be assumed that one manufacturer can maintain for long a supremacy due to superior personnel. The rival will soon appreciate the value of your president, superintendent, or manager, and make it of pecuniary advantage for him to change. Neither can an industrial concern economize by under paying its men or by continuing to operate antiquated or poorly repaired tools or equipment.

MANAGEMENT OF INDUSTRIAL AFFAIRS.

It may be asked then, how can an industrial concern effect economies that will enable it to outstrip its competitors? The answer is *efficiency*. There must be an efficient plant and equipment, efficient men and efficient management. In any works the greatest of these is *management;* for an efficient management tends to produce efficiency of plant, equipment and men.

THE NAVY COMPARED WITH AN INDUSTRIAL ORGANIZATION.

While we have no commercial rivalry in the Navy we do have a most powerful incentive to efficiency in the consciousness that in the supreme test of battle, if that should come, the most efficient fleet or ship must win. And in the maintenance of a powerful navy economy is most necessary in order to make the best possible use of the funds placed at our disposal. The demand for economy and efficiency in the Navy is as pressing as in any phase of commercial life. If the Navy is not efficient as a fighting machine then it has no reason for existence. We will assume, from its past history, that the Navy will prove its efficiency as a military instrument in time of war. If this assumption is well founded the cost of that efficiency during the period of actual hostilities will not be questioned, and need not be considered.

Ships are built, equipped and manned with an eye single to their achievement in the hour of battle, yet every patriot hopes that hour will never come.

A nation with world interests must either be in a position to wage war offensively or must be geographically isolated so that it is immune from attack. We either need no Navy or else a Navy large enough and efficient enough to be a practical guarantee of immunity from attack from any single foreign power whose interests of policy may run counter to ours.

The Navy's friends in and out of Congress seem to have adopted the policy of maintaining a Navy sufficiently large to insure peace. If this be the policy we must at all times

maintain against the hour of battle a moderate sized fleet of powerful armored ships in a state of high efficiency, and the administration of the navy becomes then a problem of industrial efficiency, measured in different units from those employed in industrial life.

Carrying the comparison a little further, the *object* of a factory is to make and sell goods. All dividends being derived from profits on manufactured articles sold, the factory site, buildings, equipment and men, are but contributory factors. The sole reason for the factory's existence is to manufacture goods to be sold at a profit. So, in the navy, the fleet, the ever-ready battleships, is the whole cause and object of the large and increasing annual appropriations.

NAVAL APPROPRIATIONS.—REASONS FOR.

Congress votes millions annually, and the people cheerfully provide them, not that there shall be maintained a navy yard here or there, nor that a few hundred or thousand workmen shall be given employment; but that an efficient fleet shall be maintained for the protection of our homes and to secure the national honor. The people expect, and have a right to demand, that every dollar voted for the Navy shall be so expended as to give the greatest return in offensive and defensive power afloat. The number of ships, their types and characteristics; the location and maintenance of navy yards; the purchase of supplies, and the decision as to whether ships shall be built or repaired in private yards or in navy yards, ought to be divorced from political considerations. These and similar questions should be decided solely by the standard of military efficiency; *i.e.*, what method will give the country the greatest number of effective ships in commission, ready at all times for instant duty, at the least expenditure of the people's money.

When a navy yard or ship outlives its usefulness it should not be perpetuated simply in the furtherance of local interests. The safety of the country, the property and lives of the people are too vitally dependent on the military efficiency of

the fleet to permit small selfish questions of politics, labor or commercial interests to affect the policy of national defense, or to determine the expenditure of funds voted for defense.

NAVAL ESTABLISHMENT.—CHANGES SUGGESTED.

If the above considerations were permitted to govern in the administration of the Navy, the maximum of military efficiency at a minimum of expenditure would result. The following simple, equitable and just arrangements are suggested as tending to this end:

Increase the enlisted artisans sufficiently so that in normal times the navy yard mechanics are enlisted men, performing shore duty, as a reward for meritorious and effective sea service. Secure the complement of artisans for any ship that has been modernized from the mechanics who did the overhaul work on her.

Base ships definitely on specified navy yards, so that their crews will know that what time the ships must spend in replenishing supplies or refitting will be at the home yard, thus enabling the men to have a certain amount of home life during their sea service.

Abolish all corps differences. Put all officers on the same basis of rank and promotion, as they now are on the same scale of pay. Let officers specialize according to their inclinations combined with special ability. After reaching a certain grade let those who specialize perform shore duty only in their particular specialty, without prejudice to their promotion. To illustrate: Take ensigns after three years' total sea service and give them graduate work in ship design and naval architecture. After the completion of this course send them to sea on first-class ships not exceeding two-fifths of the time until they reach the rank of lieutenant commander. Arrange this sea service so that these technical graduates are on board all big ships at target practice, whenever power runs are made, and during long cruises—that is, whenever the ship is likely to undergo severe stress. After reaching the

grade of lieutenant commander, they should go to sea only as special technical aids on the staffs of flag officers.

BENEFITS EXPECTED.

Some of the benefits that would accrue from the above would be:

Among officers the elimination of all corps differences. The common aim of every officer would be naval efficiency as measured by ship efficiency. There would then be but one party in the service, every member of which would be striving for the general good of the service; and an officer would be selected to do given work, solely because of his ability, and without regard to service politics.

In the navy yards all labor troubles would be avoided, because all employees would either be under military control or would belong to the classified service.

Among the enlisted men afloat the men in the higher rating would become a permanent force. They would become family men, and plant their homes in the vicinity of definite navy yards. This permanency or secured tenure, together with the extra reliability that comes with family responsibility, would immensely improve the character and increase the worth of our men.

The enlisted artisans on board ship and the enlisted artisans in the navy yard would have common interests, and could be worked together on any and all jobs, to their mutual advantage, and to economic advantage of the Government. The officers ashore and afloat would be in charge of exactly the same type of men and mechanics, so that their methods would be identical.

This identity of purpose and similarity of means would make it simple and feasible to have the senior engineer officer of a ship become in fact an assistant to the heads of the industrial divisions in the yard, in charge of all repair work on board his ship. This would give the yard department a local commissioned assistant on every ship in commission undergoing repairs, an assistant whose knowledge of the needs of his

own ship is (or ought to be) perfect, and who is more vitally interested in the work on his particular ship than any one else.

PRACTICABILITY OF PROPOSED SYSTEM.

It is realized that however desirable the above solution may be it is as yet unattainable so far as enlisting the yard mechanics is concerned. There is, however, no apparent reason why the other features should not be put into effect. A house divided againt itself is sure to fall. If personal, professional or corps ambitions are placed above military efficiency, which is the chief interest for the service good, there is sure to be internal strife and dissension which will militate greatly against general advancement. This is not theory, it is fact. Within the experience of the writer the most vicious corps strife has been absolutely abolished by welding the warring factions into a homogeneous whole, having common rights and privileges, equal pay and rank, and one purpose.

Naval officers, more truly, perhaps, than any other class of American citizens, are servants of the people. We obey without question the people's behest expressed through any Government edict. We are, it is true, the targets for press criticism; but the Navy enjoys in large measure the confidence of the people. This confidence is justified by the history and traditions of the Navy. It is not ours to define governmental policies, but to be the Nation's strong right arm in the execution of such policies. We ought nevertheless to determine and define the internal policies of the Navy. If the Government's policy is to have a fleet that is sufficient in numbers, power and efficient readiness to deter any probable enemy from precipitating hostilities, it is for the people to say through Congress how much they are willing to spend; for congress, on the advice of the Department, to determine the sizes and types of the ships needed, and the numbers possible within the appropriation. It becomes the province of the Department and the officers of the Navy to expend the people's money to the best advantage. The Department should not be hampered by ulterior questions or restrictive legislation. The questions

the people should ask and demand the answer to, are: "Have we the most efficient fleet possible for the money expended? If not, why not?" "Who is responsible?" "What is necessary to be done in order to remedy the fault and prevent its recurrence?"

EFFICIENCY OF SHIP ORGANIZATION.

It is believed that the ships in commission are in a state of efficient preparedness. It is also believed that the actual operating expenses of the fleet, *i. e.*, coal, oil and consumable supplies expended per mile steamed, are in the neighborhood of what they should be, with fair prospects of continual improvement, due to the system of competitive rivalry recently inaugurated by the Department.

NAVY-YARD ORGANIZATION.

Such a satisfactory and hopeful condition does not now exist in regard to the maintenance of the fleet. The overhaul and repair bills are larger than they should be. This is due to many contributory causes. Political and labor questions cause ships to be built in navy yards when they could be more economically and expeditiously built in private yards. They determine the policy that a fixed number of men must be kept on the pay roll at each navy yard. This policy gives rise to expenditures on obsolete craft out of all proportion to their repaired value. It also tends to limit the repair work that may be done by the ship's force while the ship is at a navy yard. In order to provide work for the navy yard force it has heretofore been customary to authorize many alterations which, however desirable, were in no sense necessary. Happily, such alterations are becoming less frequent. Questions of navy yard management, due to politics and service differences, affect injuriously economical administration.

FORMER PREPONDERANCE OF MILITARY TRAINING.

Ship-board management has heretofore largely neglected the industrial side in a vain endeavor to make enlisted mechanics and unskilled laborers useful as military men. Happily, now that steaming competition is linked with gunnery com-

petition, those in authority are beginning to realize that a mechanic's highest military value lies in his ability to keep the machinery in a state of efficency, rather than his ability to perform indifferently half a dozen so-called military exercises.

MECHANICAL EFFICIENCY ESSENTIAL TO MILITARY EFFICIENCY.—SHIPS TO MAKE REPAIRS.

Commanding officers now realize, as they never have before, the absolute necessity for engineering efficiency as a means toward military efficiency; there is promise in the near future:

That artisans will be called on for a minimum of so-called military drills, and that the rest periods of the unskilled laborers of the engineers division will be broken with less frequency.

That the responsible officers will be allowed regular periods for overhaul and adjustment of machinery.

That the paramount duty of officers and men performing engineering duty will be to keep machinery in an efficient condition and to operate it economically.

That in the performance of this duty they will be given the intelligent support of those in authority, and be exempt from many of the vexatious interferences to which they were formerly subjected.

If these promises become wholly realized, there will be no valid reason why the ship's force cannot and *should* not make *all* the current repairs on board every large ship in commission.

PROPER FUNCTION OF NAVY YARDS.

It is the opinion of the writer, corroborated by that of many able officers in the service, that the proper functions of a navy yard are as follows:

To furnish docks and docking facilities.

To be a supply depot for munitions of war and for the replenishing of consumable stores.

To furnish rendezvous for the fleet where the mechanical force can make all repairs within its capacity while the deck force is doing such unskilled labor as scraping and painting

the ship's hull and getting stores and munitions aboard, and from which all men may be granted shore liberty and short leaves of absence.

To furnish shops and mechanics to make such repairs as are beyond the capacity of the ship's force and plant.

To do all work in modernizing ships.

To manufacture and assist in installing the parts that go to make up new installations or alterations on ships in commission.

To manufacture such articles for the Navy as are by their nature or the use to which they are to be put so special that they can be more economically or satisfactorily manufactured in a navy yard than in a commercial plant.

If these are accepted as the only proper functions of a navy yard, then by making each yard a manufacturing plant for certain specific articles needed by the Navy — for instance, chain, rope and small pumps at Boston; ship's boats at League Island; ship's furniture at Norfolk; steam launch engines at Portsmouth; special instruments and appliances at New York, etc. — and by keeping in constant employ at each navy yard a force which is a definite percentage in excess of that required for the specific manufacture, say 25 per cent. excess, it will be possible to be always ahead on the manufactured articles sufficiently to justify withdrawing mechanics from manufacturing and employing them on such repair work on ships in commission as is beyond the capacity of the ship's force and plant.

DETERIORATION OF SHIP'S MACHINERY WHEN DEPENDENT ON NAVY YARDS.

Whether the functions of the navy yard are delimited by official orders or not, it will be scarcely questioned that all work possible ought to be done by the ship's force, thus saving to the Government the large labor cost involved in all navy yard work. Whatever the figure of the monetary economy thus shown, it is insignificant compared with the military value of constant readiness. To quote a Department publication:

"Two large, high-powered vessels of the service require such extensive repairs in the engineer department as to render it necessary in the near future to lay them up for repairs extending over a period of several months. Until these repairs are completed, these ships are without strategical value as a component part of the fleet. The present condition of the machinery of these vessels can best be described as generally run down."

This shows an extreme case; but, allowing for differences in times to repair, this is true of every vessel in the Navy that does not make *all* repairs with her own force. The last sentence of the quotation is a Departmental expression of a fact well known to each of us, namely, that the "run down" condition of the machinery of a vessel is largely due to the accumulation of a multiplicity of small items, each in itself of comparatively little importance. It is believed to be a statement of fact that the average request for navy yard repairs, when carefully analyzed, shows that nearly all the work could have been done by the ship's force, had the repair for each breakdown been made an emergency operation and proceeded with in a thorough-going way. Effecting a repair immediately after a breakdown secures two results; it strengthens the entire structure, thus preventing other breakdowns, and has a most beneficial educational and moral effect on the mechanical personnel.

REPAIRS SHOULD BE MADE PROMPTLY.

The repair of each breakdown ought to be undertaken at once and prosecuted day and night to its completion. In this way the men are trained in the correct habit of thought and the ship is as nearly as possible always ready for any duty that may be assigned.

STANDARD ORGANIZATION FOR THE ENGINEER DIVISION.

Any officer of sufficient experience can, and most officers do, maintain their ships in a high state of efficiency. Each officer is valuable by virtue of his knowledge and experience;

but when, in natural course of rotation, he is detached from his ship, he usually leaves no complete data, no standard organization or method that will "carry on." After a time, and usually a very short time, the methods of the experienced officer give place to the spirit of change that possesses us all, and the ship that had been able to bar out the navy yard workman now welcomes him as a savior. This is due to our lack of system, to the fact that we have neither standard organization nor scientific management.

The subject of organization and management will be divided into two parts: First, a standard organization will be offered, holding to our present system; and, second, a detailed system will be proposed to be superposed on the standard organization. The first may be accepted and used without reference to what comes after. The effort will be made, however, to show that detailed system management is the logical development of standardized organization.

It is unnecessary to argue the desirability of uniform organization or to deal in general theories as to the best features of organization and administration.

An effort is made to present to the service a concrete organization, so general and flexible that it may be easily adapted to any ship, and at the same time in sufficient detail that it will serve the purpose for which individual ship organizations are worked out.

No originality is claimed for anything offered. In fact, free use has been made of all available data on the subject. After the bills were practically completed the writer received a copy of the Commander-in-Chief's report on standard bills, and, finding in it a recommendation that all auxiliary watch and station men be segregated and placed on a separate bill, he had to modify the completed bills in order to embody this excellent and, to him, new idea.

The following bills are proposed:

(a) A general cleaning station and repair station bill.

(b) A general organization bill, in four sections, which includes all the steaming watch men.

(c) A combined organization and station bill for the fifth section, comprised of all men who stand auxiliary watches, or perform auxiliary station duties.

(d) A general action bill.

(e) A coaling bill.

There are also proposed the following forms:

(f) A tool check board.

(g) Ice-machine record.

(h) Evaporator record.

(i) Fireroom operation record.

(j) Foundry order.

(k) Work order, chiefly used in ordering work in power shop.

EXPLANATION OF BILLS.

(a) *The cleaning bill* is made for the ideal largest battleship. No names appear on it. Men are stationed by ship's numbers at the principal and auxiliary stations. A definite number of men to each compartment or auxiliary station. This bill gives the chiefs of cleaning and repair stations. It provides that all men on any cleaning station are to be used for any repair work on that station. It shows that all men work on prescribed stations under the direction of definite petty officers. It provides that the senior petty officer in a compartment or on a station shall muster, account for and direct the work of all men in that compartment or on that station. It also contains certain cautions as to casualties and the safety of the men.

The system of numbering is as follows: Each man is designated by four characters, either four figures, or three figures and a capital letter, except in the case of special mechanics, where two letters are used in conjunction with three figures. Four figures denote men below the rating of petty officers. A capital letter preceding the figure indicates the highest grade in that particular rating. If the letter follows the figures the man is a petty officer, though not of the highest rating. Arbitrary division has been made so

that a man's rating is immediately known from his number. It is assumed that the engineer division shall be the sixth division on the ship; therefore the first figure is always a "6." All principal station men have 1, 2, 3 or 4, as the second figure in their numbers. So far as possible the last figure is made to indicate the compartment to which a man is assigned for cleaning. This is rigorously true in regard to non-rated men on fireroom stations and in the engine rooms.

(b) *The organization bill* gives the columns for the entry of such stations and duties as affect the men in their ship-board relationship. There are four separate, small bills, each called a watch bill, and intended to station the men for steaming watches under normal cruising conditions when one-fourth of the complement is sufficient. Each watch is divided into three sections, so that when it becomes necessary to go into a "watch in three" the men stand watch by sections. Thus the first section of all four watches would be the first watch, etc. It will be necessary to shift from "watch in four" to "watch in three" in the middle of a watch period so as to avoid having some men stand eight hours continuously. When steaming at full power and it becomes necessary to stand "watch and watch" all men on 1st and 3d watch bills constitute the first watch, and all men on the 2d and 4th bills, the second watch.

(c) *The combined organization and auxiliary station bill* segregates all special mechanics, yeomen, toolroom and store-room keepers, washroom keepers, passage cleaners, steam launch men, *permanent* auxiliary machinery details, water tenders in charge of auxiliary watches, messmen, and the chief water tender in charge of bunkers and chief machinist mate in charge of the power shop. It contains all the columns on the organization bill and in addition has a column that gives the duty station of each man. The second figure of all numbers on this bill is a "5." As far as possible the 3d, or last, figure indicates the watch, when firemen or coal passers are detailed for the auxiliary fireroom watch. Thus: 651-W.

would be in charge of the watch, made up of firemen and coal passers detailed from the first steaming section, etc.

(d) *The general action bill* stations all men, in a general way at general quarters, fire quarters and collision drill. It will be noted that all the general requirements for these drills are provided for, that all men have the same stations, and as far as possible the same duties at all drills. All coal passers are assigned to the powder division. Those on watch when general quarters are sounded go to their powder-division station without waiting to be relieved. It probably will happen on most ships that all coal passers will not be needed in the powder division. This condition is met by passing the word that coal passers on watch remain on duty in the firerooms.

(e) *The coaling bill,* like all the other bills, is general. It is believed, however, that it is based on correct principles, and it is known from experience that it serves its purpose in making all preparations for coaling and of insuring that men are at every bunker ready to receive and trim coal. It will be necessary on each ship to assign the officer in charge of weighing and tallying party, and to make a diagrammatic tracing giving bunker numbers and capacities, together with frame location of chutes. From this tracing numerous blue prints are made. Before coaling is begun the senior engineer officer makes, or has made, a careful inspection of all bunkers. On two of these blue prints he enters in pencil the amount of coal that each bunker will require to fill it. One of these, sent to the executive officer, serves to inform him how much coal is needed, and where, so that he knows where to place colliers or lighters, and is able to properly dispose the deck force. The other print enables the senior engineer officer to instruct the water tenders as to the amount of coal each bunker will take, and to make proper disposition of the men.

Experience proves that the water tenders, firemen and coal passers can stow and trim the coal as fast as it can be taken on board. It is also a fact that commissioned officers and

midshipmen are more valuable as officers in charge of coaling than are warrant officers. For this reason it is a good policy to carry on general repair and overhaul work during coaling, utilizing for this purpose warrant officers and all artisans.

By regarding the coaling bill as a means of starting operations only, and reserving and exercising the right to shift men from bunker to bunker where their services are most needed, and by the senior engineer officer properly directing which bunkers coal shall be dumped into, so that only one bunker of a group is at the close-trimming stage at one time, it is an easy matter to trim coal just as fast as it is taken on board, until all bunkers except the last are filled. At this last bunker, if men are worked in relays, and proper attention is given to the trimming, the rate of trimming will not become the limiting condition until the bunker is within ten tons of full. These last tons may be dumped on deck, and be trimmed while the deck force is unshipping coaling gear.

STANDARD BILLS.

It is recommended that the above described bills be supplied by the Bureau in blue-print form to each ship. If this be done the method of adapting them to a particular ship will be as follows: Take the cleaning bill and allowed complement. Determine what compartments and stations provided for on the bill do not exist on the ship. Blank these off on the cleaning bill. Then decide how many men and what rates can be assigned to auxiliary stations. Divide the remaining allowance up between the compartments of the principal stations. From this preliminary work it is known how many chief water tenders, water tenders, firemen, first and second class, and coal passers are available for each fireroom, for instance. Beginning with the first numbers in each rating check off on the cleaning bill the ship's numbers for which there are men available in the complement. Blank out all other numbers in that rating for that station. Do the same for other ratings and all other compartments and stations. This will leave a cleaning bill with ship's

numbers equal to the number of men carried on the complement. Now check those numbers that remain on the cleaning bill wherever found on the organization bill and auxiliary station and watch bill. Blank out all numbers not checked. Finally, against all remaining numbers, enter names and other data required. This will insure every ship having exactly the same assignment of men to stations and compartments that are common.

EXPLANATIONS OF FORMS.

A tool check board is to be furnished. It contains the names and sizes of all tools ordinarily furnished to battleships that are small enough to be lost or carried away. A pin or hook is provided for each tool of every size. If the Bureau should furnish several prints of this board to each ship these may be made as required, using any kind of soft wood that may be available. Simply paste a blue print on a board and screw in hooks where indicated.

If the Bureau would have stamped out, at a navy yard, small brass tool checks corresponding to the ship's numbers on the cleaning bill above mentioned, then the senior engineer officer, by notifying the Bureau what numbers are utilized on his ship, could get tool checks for his entire force.

By issuing a certain number of checks — say ten — to each man, and issuing tools only in exchange for checks, you have at all times a complete record of all tools. The number of checks on a pin show exactly the number of that particular tool out of the storeroom at any time. This number plus the number of the same tool in the bin ought to equal the ship's allowance. The ship's number on the checks on the pin show exactly what men have that particular kind of tool out.

By requiring each man to "cash in" all tools and so redeem his checks before allowing him to leave the ship on liberty, the greatest care is exercised by the men, with the result that the tool loss practically disappears.

If a job is under way, and it becomes necessary to change

mechanics and men, it may happen that it is inconvenient to return the tools. In this case the relieving men make careful inventory of the tools in use, go to tool-room window, in company with the men who are being relieved, and there effect an exchange of checks, so that the men who actually have the checks are charged with them by their own checks.

Experience proves that commanding officers will enforce the order in regard to "cashing in" tools before going on liberty or leaving the ship. It is also the experience of several officers in the service that the constant use of this system prevents practically all loss of tools. One ship shows a loss of only 53 cents for a year.

Other forms are proposed for use as required. The *foundry order* and the *power shop* or *work order* are each double forms. The first part is an order to do a certain thing, and the other part is a record of work, time and material employed in doing it.

The forms are all of uniform size. The idea is that blocks of these forms will be supplied by the Bureau to each ship; then, by having a small box with as many compartments as there are different kinds of forms, each labeled, it will be easy and convenient to use the proper form, and save clerical work. The operation record forms will be kept in the firerooms, evaporator room, or ice-machine compartment, as the case may be; but when they are filled out they come to the office and must be filed.

METHOD OF CARRYING ON WORK.

In the administration of the engineer department with the foregoing organization and forms, under the so-called military management, efficiency is obtained by covering all the repair stations and work with the complement of chief machinists and machinists. These officers are skilled mechanics and should be utilized as leading men, foremen and directors of the work for which they are best fitted.

The repair of every breakdown, whatever its nature, should be regarded as emergency work, to be undertaken immediately

and prosecuted, day and night, to *its* completion. This work should be done under the personal supervision of the warrant officers of the station. In order that warrant officers may be available as superintendents of all repair and overhaul work they should be exempt from watch duty as far as is consistent with the number of other officers assigned to engineering duty. Watches should be kept by commissioned officers. If there are not enough such officers to stand all the watches midshipmen should, under the supervision of the senior engineer officer, stand the day watches. If there are a sufficient number of commissioned officers they should stand the watches while midshipmen stand junior-officer watches. The watch officer should spend most of his time in the firerooms, as it is from the boiler operation and management that greatest improvement may be expected.

SUPERVISION OF WORK.—ASSIGNMENT OF OFFICERS.

Both warrant officers and midshipmen should be assigned to the various divisional duties; but the warrant officers should never be required to perform these divisional duties when their services as repair officers are necessary.

In addition to the principal division work, all inspections for cleanliness, care and preservation of hull and machinery, condition of safety appliances, together with the supervision of stores and tools, their condition, availability and sufficiency, their requisitioning and issue, should be distributed among the commissioned officers and midshipmen performing engineering duty.

In addition to the various duties outlined above for midshipmen they should, during their tour of duty, be required to actually set up and do various kinds of work on power tools, always under the direction of a skilled mechanic; to operate and overhaul steam-launch machinery; to handle the main engine in answer to signals from the bridge; to be on station every time the ship is got underway or anchored, and always at securing machinery. They should be required to observe and take part in all repair work, general overhaul and

adjustment operations; to conduct or assist in all types of scientific tests and operations, such as valve setting, determination of clearances, testing feed water, analysing flue gases, making boiler measurements, inspections, and determining whether or not boiler water is corrosive. They should take indicator cards and work them up. Their journals should contain description of all work that they have done or observed, together with a log of all important happenings occurring during their watches. They should contain sketches and descriptions of particular installations, and of all changes and new installations, rather than information copied from specifications or from the approved drawings of the ship or machinery.

Midshipmen should stand regular watches in port, in charge of the auxiliary watch. If they are alert and attentive the reduced coal consumption due to their presence will be very marked. The discipline of the watch will be improved. The midshipmen will have a real responsibility, will thoroughly learn all the auxiliary machinery, and will gain an intimate knowledge of the men on watch.

Every part of the plant should be covered by an officer, officially designated, either a commissioned officer or a midshipman, who reports verbally or in writing, to the senior engineer officer, the results of his daily and periodical inspections.

ADVANTAGES OF UNIFORMITY ON ALL SHIPS.

Thus far nothing has been advanced that is not perfectly familiar to all officers. The aim has been to emphasize the desirability of having uniform organization, with standard bills and forms, and a standardized method of performing duty. An effort has been made to supply ready to hand such bills and the more important forms, so that all ships may, at a minimum of trouble and work, go into standardized conditions. No claim is made for the superiority of the bills, but it is believed that they are based on correct principles, and are so flexible that they can be readily adapted to all ships.

It is evident that any fairly good organization, that is stand-

ard for all ships, is better than the best non-uniform, individual ship organization can possibly be, so far as the entire fleet is concerned.

THE APPLICATION OF SCIENTIFIC MANAGEMENT.

In what follows the writer does not expect that the service, or any officer even, will agree with him entirely. The hope is to present arguments showing the need for greater detailed management, to describe some of the methods and mechanism used in securing and recording the necessary information, and to give a brief outline of a proposed system of management. It is not suggested that any ship undertake the installation of any part of the mechanism necessary for this type of management until it has been demonstrated by practical application on board a ship that it is workable and that it gives returns commensurate with the extra work and trouble inseparable from getting at details. The writer expects to have the opportunity to install this system of management on one of the new ships, and hopes to be able at the end of a cruise to show a concrete example of operating under the system of management outlined below. It is to be expected that there will be a great difference between the present tentative scheme and the future scheme rounded out in practical use.

THE MAN AND HIS WORK TO HARMONIZE.

The object of all management should be to increase output, both as to quantity and quality; to economize by eliminating preventable wastes of time, material and equipment; to reduce labor costs and fixed charges by increasing laborer and machine efficiency. To do these things, or any of them, it becomes necessary to standardize; to specify what the workman is to do, and how, and in what time he is to do it; to so sub-divide work that every workman is doing the highest grade of work of which he is capable (to put a machinist to clean a feed tank is like putting a race horse to a trucker's cart), and to see that a workman does a fair day's work for his day's pay. These

things can be accomplished only by assigning work to men individually. To do this we must know what is to be done, how it should be done, and the time it ought to take. This information is secured by analyzing every job, determining the operations necessary, the men and machines that can best perform these operations, and the time required. If these elements had to be separately determined for every job that comes up the work would be colossal and there would be little or no gain.

Fortunately, by studying the men and machines and by a series of time studies of the elemental operations, it becomes possible to build each job up synthetically from the elementary data collected and recorded. In this way each job can be planned in advance. The work can be apportioned to the men best fitted for the execution of any particular part. Definite detailed instructions as to methods of doing the work and the time allowed can be provided. All this planning and time setting, as well as all supervision, instruction and inspection, would be done by the officers and highly-skilled mechanics, as it should be. The man who actually does the work should not have to plan it, or to provide his tools and equipment, any more than the soldier who fights the battle has to plan the campaign or equip the army.

INCENTIVES TO EFFECTIVE WORK.

Men usually work only from necessity. To secure a full day's work there must be some reward that appeals to the individual man. Perhaps the most efficacious system of reward in civil life is the payment of a money bonus for efficient performance. If a man follows instructions and performs the task set him he is credited with a percentage increase of time, varying from 30 per cent. to 100 per cent. at full pay. In any military service the hope of commendation or the fear of censure operates as an incentive. In addition, so far as the enlisted men are concerned, a powerful incentive to honest work lies in the assurance of just treatment — a square deal — and sure reward in the way of privileges and of promotion. As

a corollary, the men must feel certain that every man who "soldiers" will be punished. No other one thing within my experience so demoralizes the enlisted man as his knowledge that certain men on board — "coffee coolers," to use the men's vernacular — act as if the Government owed them a living, and are not punished therefor. If there are 20 per cent. of such men in a force, by discharging the 20 per cent. we can and do get more work from the remaining 80 per cent. than it was possible before to get from the entire force. At the same time the happiness and contentment of the men are increased.

If men are assigned individually to work and their individual records are kept, and if all promotions and disratings are based on these records so that men are rated and disrated on their efficiency records only, we have in effect an actual *bonus* system similar to the best used in any manufacturing plant ashore.

DETAILED ARRANGEMENTS NECESSARY.

The details of information, records and mechanism needed for the scientific management of the engineer department on board ship are somewhat as follows:

Experimentally determined time studies of the elements that make up the usual routine overhaul and repair work.

A knowledge of the capability of the men.

A standard arrangement of tool rooms and storerooms, with standard method of issuing and accounting for tools and stores.

A familiarity with shop processes and an intimate knowledge of the power tools.

Standard forms for ordering work, issuing tools and stores, assigning tasks to men, and recording work and the results of tests and inspections.

Suitable mechanisms; such as cabinets, racks, file cases, storeroom and tool-room arrangements for filing the various forms, etc.

Classification of installed machinery, parts, tools, stores and operations, so that things can be specified by a symbol consisting of a few characters instead of by a long written legend.

This is essential both for the sake of reducing clerical work and in order that all data and records may be filed alphabetically, as well as that stores and tools may be arranged alphabetically by symbol.

In the description that follows it appears best to explain the mechanism first, then the forms and their use, and finally to show how they all dovetail together and provide for all the requirements of systematic management.

MECHANISM FOR MANAGEMENT.

In the office are the following: A tickler cabinet containing twelve drawers. Each drawer to contain thirty-one manila folders, numbered consecutively from 1 to 31, inclusive. A small open-ended cabinet to contain a block of each kind of form used, a separate labeled compartment for each kind of form. A board or cabinet to hold the individual work orders, styled a Work Order Cabinet. In the engine room and at other convenient and desirable places in the department, a small department mail box. In the tool room and storerooms proper bin and drawer arrangements for holding all tools and stores. These bins should be of a standard unit size with standard multiples, so that the space needed for a given kind of stores may be increased by the process of moving other stores along. The relative arrangement of stores remaining always the same. In addition there must be some filing drawers in the office, and some "tote boxes" in tool rooms and storerooms.

FORMS.

The forms needed are as follows:

(a) *Two tickler blanks.* One of these contains a space to be filled in by an officer or man reporting any fault or the necessity for any work; a space for the engineer officer to designate the person who is to do the work; and, finally, a space to be filled in by the person who did the work, certifying to its completion and giving the time of completion. The other tickler form has printed the names of the most usual operations, and on the other half the names of the things requiring

most frequent attention. Each name has a little blank square opposite it. The senior engineer uses this form to order inspections, tests and standard work. There is a place for the entry of certificate of completion, with time.

(b) *Individual work orders;* printed on white and brown paper, gummed alternate colors in pads with carbon between different colors. These are used to assign specific work to an individual man. They will be explained fully in discussing their use.

(c) *Instruction cards;* used in detailing sequence of operations and methods of performing work.

(d) *Tool lists;* used in providing necessary tools for a given job, goes with the tools as an inventory, is returned with them to enable tool-room attendant to promptly replace and account for tools.

(e) *Stores issue;* used to provide supplies for each separate job, so that stores are charged to proper expenditure. They are made in duplicate, so as to serve as vouchers both for the paymaster's representative and for senior engineer officer.

(f) *Parts issue;* used for ordering or assigning spare parts to a particular job, in same way stores issues are used.

(g) *Delinquent report.* Convenient form for petty officers in charge of compartments to use in reporting men who fail to turn to on time.

(h) *Tool report.* Convenient form for tool-room keepers' use in reporting damage to tools or loss of tools.

(j) *Tool receipt,* in duplicate; to be used when tools are issued to machinists or special mechanics who have lockers or chests in which to keep tools and who have need of many tools. One copy serves as an inventory and accompanies the tools, the other copy serves as a voucher to the tool-room keeper.

(l) *Perpetual inventory.* A form attached to each bin containing stores, on which is entered quantities received and issued, with balance by difference, so that a glance tells the quantity on hand at any instant. From it can be seen the average expenditure.

(m) *Machinery repair record,* one card for each machine on board. This record will show all repair work to any part, the time taken to repair it, and the stores or spare parts used.

In order to use the forms needed for this detailed management without involving a prohibitive amount of clerical work it is necessary to classify the installed machinery, the parts, tools, stores and operations, so that anything may be represented symbolically by a very few characters.

To facilitate work it is also essential to have standardized storerooms and tool-room arrangements.

THE ADMINISTRATION OF THE ENGINEER DEPARTMENT UNDER THE PROPOSED SYSTEM.

INSPECTIONS.

Divide the entire plant and equipment up for inspection purposes between the commissioned officers and midshipmen available. It is better to functionalize this inspection duty; that is, assign one officer to inspect hull and structure; another to inspect safety appliances, including watertight doors; a third to inspect all steam, exhaust and feed-pipe lines; a fourth to inspect boilers and boiler fittings, including furnaces, for condition and airtightness; a fifth to inspect all auxiliary machinery; a sixth to inspect slip joints, reducing valves, traps, etc.; a seventh to inspect heaters, heating system, and evaporators and distillers; an eighth to inspect valves, studs, bolts and nuts, and other securing devices installed; and so on, utilizing every officer as a functional inspector, being sure to cover every part of the plant with the officers available. In the order setting forth the specific things different officers are to cover, make it emphatic that, in addition to their specialized inspection duties, they are to report any and everything they see that is not exactly as it should be.

REPORTING DEFECTS.

Issue the following general order, and see to it that every officer and man in the engineer's force understands and obeys it:

"Whenever any officer or man observes any part of the ship or machinery within the machinery space or under the cognizance of the senior engineer officer dirty or out of order in any respect, or sees any steam or other leak, or notes any machine or part running noisily, or observes any waste of fuel, oil or stores, he shall immediately report the fact in writing, in the following manner: If an officer, or chief petty officer, he shall enter on a tickler blank, date, fault and location, initial and mail; unless the fault requires immediate righting, in which case, if he have the authority, he shall direct the necessary procedure; otherwise he shall also immediately report verbally to the senior engineer officer. If a petty officer or man, he shall report verbally to the officer of the watch or other officer most convenient, whose procedure shall be the same as if he had discovered the fault originally."

In this way we not only have all parts of the plant covered daily by a functionalized inspector, but several official inspectors go over the same ground and check one another; and in addition we are utilizing the eyes of every man in the force to ferret out, locate and report faults. These inspections will tend to eliminate faults, will train all hands to careful observation, and will make the men feel that they are essential to the proper administration of the department.

METHOD OF MAKING REPORTS.

In order to systematize the making of the above inspection reports and to insure that they reach the senior engineer promptly, and further, to assist him in directing the correction of the faults and to enable him to receive reports of the corrections we use the following: tickler cabinet, tickler forms, departmental mail and mail boxes. Locate in engine room, in machine shop and in central fireroom, a little mail box capable of taking, without folding, the tickler form.. In these same places keep a block of the tickler blanks. Assign a bright, active coal passer as department mail boy. At stated times this mail boy makes the rounds of the department, delivering to the proper persons any memorandum found in the tickler

folder for the given date, and at the same time he collects all tickler blanks found in the various department mail boxes. These the senior engineer officer looks over and, in proper spaces, enters initial of person who is to do the work, together with time when work is to be finished. If the work is to be done that day the tickler blank with the second entry remains in folder of same number as date, and is delivered by mail boy to proper person, who, when work is completed, certifies to the completion by entering time and initialing. If the work is to be done on some subsequent day, the form with second entry is put in folder of same number as date on which work is to be done, and comes out on that day by regular mail routine. By means of this same tickler form and the special one the senior engineer can order all the work, inspections and tests of the department. When the work is executed these same forms come back to him, and serve both as a certificate of the responsible person and as a record of performance. Since there is a tickler folder for every day in the year, each folder, after it ceases to be active, on the date corresponding to its number, serves as a file record of everything that was done on that date. It may be retained as a file record until nearly the same date in the next year.

DISTRIBUTION OF WORK.

The information collected and data recorded through the agency of the department mail and tickler equipment puts us in a position to know beforehand what work is to be done in a given compartment on the following day. From the cleaning bill we know what men are available on a given station. From the report of the mechanic or chief petty officer we know what work is needed that might not be discovered through inspection, as, for instance, the necessity for cleaning a boiler or grinding in certain valves. After working hours, or at any time when they are not otherwise engaged, the officers, warrant officers and mechanics and chief petty officers in charge of stations come together, and, from the information mentioned above, distribute the work that is to be

done the following day in each compartment to the men available. Each chief petty officer will know his own men intimately, and will be able to assign to each man the work for which he is best fitted. The officers and artisans will be able, between them, to make a fairly good estimate of time required for a given job, and they will be able to determine very accurately the stores or spare parts needed, and to specify the tools required.

WORK ORDERS.

They will fill in the individual work orders, one for each man for each separate job, giving the time allowed and indicating whether or not a tool list, stores issue, parts issue or instruction card accompanies the work order. If any or all of these do, they will make them out. All of these forms filled in for a given job will be identified by the same classified symbols, both machine and part. The work order and stores issue will be in duplicate. All stores issues and parts issues will be placed before the senior engineer officer for his approval by initialing. This enables him to keep a personal supervision over all expenditures.

PREPARATIONS FOR EXECUTION OF WORK ORDERS.

These filled-in work orders, with the accompanying necessary lists, are filed in the pigeon holes of the work-order cabinet. This cabinet has a pigeon hole for every man in the department and is a fixture in the office. It is duplicated in sections, so that each chief petty officer has a section with as many pigeon holes as he has men.

Before working hours the tool messenger gets all tool lists, secures from the petty officers or men who are going to use the tools the correct number of tool checks, and secures from the tool room, on check, the tools specified on each tool list. These tools, together with the tool list, are put in a tote box, and the messenger places them at the work ready for the mechanic.

At the same time the stores messenger gets all the stores

issues and goes to the storeroom, where the paymaster's representative issues the stores on spare parts called for by each issue as a separate order. These are put up in tote boxes or in packages, and the messenger takes them, together with the duplicate issue, to the place where they are to be used.

Just before "turn to" each chief petty officer in charge of a compartment, repair or overhaul work, comes to the office, takes the original (white) work orders for all the men under his control, together with the instruction cards, and puts them in the pigeon holes of his section of works-bulletin cabinet, the orders being arranged in the order in which the work is to be done.

EXECUTION OF ORDERS.

Thus at "turn to" all tools, stores and spare parts are already at hand; the men report on station; each gets the written order for the first job he is to do, together with instructions telling him how to do it, and specifying the time allowed. The man does the work ordered in accordance with the instructions. When he completes his job he presents his work order to the designated inspector, who *must* at once inspect the work. If it is satisfactory the inspector enters clock time of completion, with units of time taken, and certifies that the work is satisfactory by initialing. If the work is not satisfactory the inspector carefully instructs the man and sees that he performs the work in the correct manner and that the final result is satisfactory. The inspector enters on the back of the work order any comment he cares to make in regard to unsatisfactory work. The man now takes the completed work order, places it in the back of the particular pigeon hole assigned to him, and gets the next work order from the front, and so on.

At any time during the working day an inspection of the works-bulletin cabinet in the office will show the senior engineer officer what work is being done (indicated by brown cards) and what work is planned but not yet undertaken (indicated by white cards).

COMPLETION OF WORK.

When a job is completed the tool messenger takes the tools and tool list, in the tote box, back to the tool room, receives the tool checks on which the tools were drawn, and returns them to their owner. If all tools are returned in good condition the tool-room keeper releases all checks; otherwise he retains checks for tools not returned or damaged, and forwards to the office a tool report.

At the same time the stores messenger takes any left-over stores, together with the stores issue, and returns them to the storeroom, where, in his presence, the paymaster's representative enters amounts returned, and the messenger initials after "stores returned by." At his convenience the paymaster's representative enters on the original the stores returned, and then, by difference, enters on both original and duplicate "stores expended." The original serves as the paymaster's voucher, from which the representative corrects his perpetual inventory on the bins affected. The duplicate, when prices are entered and totaled, is forwarded to the senior engineer officer.

At the end of the half day or day, or at any other convenient time, the chief petty officer in charge of work returns the executed work orders to the office, where one of the office men corrects the duplicate (brown) work orders to correspond with the originals (white) returned from working stations.

These work orders are now filed, the original (white) under the man's number, the duplicate (brown) under the machine symbol.

RECORDS.

From the first file we have a complete record of every man. All ratings and disratings are to be based on this record. From the file we are also able to collect data as to the time required for specified work.

From the second file and the duplicate stores issues returned by the paymaster's representative we get all the information needed for complete entry of labor, time, and material cost on the machinery repair-record cards.

This data is entered on the machinery-repair record by the yeoman or other office assistant. There is a space on the work orders and stores issues for a certificate that they have been so entered. The senior engineer officer, or an authorized assistant, glances over these original records and destroys those that have been entered. If he has reason to suspect that the clerk is checking either the work orders or stores issues as being entered before they are actually entered on the machinery repair records, it is very easy to check him up by selecting a few of these cards at random.

INSPECTIONS.

Instruction cards which require inspections that intervene between operations will be submitted to the senior engineer officer, so that he can assign proper inspectors. He will be governed by the importance of the work in his selection of inspectors. Midshipmen and warrant officers may be assigned to make all inspections on auxiliary machinery other than main engines and boiler dependencies. Warrant officers and commissioned officers should be assigned to make all inspections of main engine and boiler dependencies and the inspection of main engines and boilers, except the final inspection before closing boilers, turbines or cylinders. These excepted cases should be inspected by the senior engineer officer. He should also be the inspector whenever a pressure test is applied.

The instruction card will make it plain that an inspection call is an operation which must be performed by the person indicated before the next operation can be begun. In this way all responsibility will be fixed and personal. There can be no closing of cylinders, for instance, with tools in them, except by the responsible officer both neglecting his duty and certifying that he has done work which has not been done.

PROPER PLANNING NECESSARY.

To the foregoing it may be objected that we are going into too great detail, and that a great deal of planning becomes necessary, as well as considerable clerical work. The answer

is that no definite, accurate information is to be had without getting at details, and that usually the difference between success and failure lies in some detail. We know that planning of some kind is necessary under any system. A plant may run for a time without any system of planning, but this is a matter of luck. If there be even a pretense of management there must be some planning, some thinking. Usually, however, under most systems, the planning, if it may be called planning,—at any rate the determination of the method of doing a thing, and the time necessary to do it,—are left to the men least qualified to plan and most interested in doing a minimum of work in a maximum time.

Under the system of management here proposed all the planning, time setting, inspecting and recording are in the hands of men best qualified therefor and most interested in getting results. Under this system every officer, petty officer and mechanic is trained to observe accurately, to inspect carefully, to plan in detail, to study every job that comes within his purview, and to instruct concisely and definitely. In short, all officers and petty officers must soon become proficient estimators and time setters, while the men learn habits of obedience and promptness, and become accurate workmen, because of constant and rigid inspection of all they do.

RESPONSIBILITY.

The system definitely fixes responsibility. It compels the management to do its part, leaving nothing to chance. It insures the equal employment of all men, each in the work he is best fitted to do. It keeps track of all tools and stores. It gives us a record of the employment of the personnel and a history of all the machinery repairs with their labor, time and material cost.

These things cannot be accomplished without paying a price, but the results are well worth the price. By means of the symbolic classifications a few characters are made to take the place of lengthy legends. This tends to reduce the clerical work; it is further reduced by having appropriate forms. All

the necessary writing is done piecemeal, and is of such a simple nature that any one, who can write legibly and read symbols, may be trained to do any part of the clerical work.

In line with shortening the writing the senior engineer officer should assign a letter or letters as an official signature to every officer or man who, by the system, is required to sign any of the forms, the men having most signing to do being given the shortest, easiest signatures.

Men performing semi-permanent duty, as auxiliary watch standing, cooking, or men in confinement, or on the sick list, will be accounted for on the work orders. By using date stamps to indicate times of beginning and completing any assignment, and having rubber stamps for the frequently recurring routine employments, nothing will have to be written on these cards except the ship's number of the men.

ESTIMATING TIME.

One of the most important, and by far the most difficult thing in connection with the work order and instruction card is correct time setting. Naturally, at first, our standard times will of necessity be estimated times. Even so, they will be nearer the truth than are times determined, as now, by the man who is doing the work, and who consciously or unconsciously lives up to the adage that "a wise man never works himself out of a job." Under scientific management those times will have to be estimated, the estimates being based, however, on the experience and knowledge of the officers and leading petty officers, and most skilled mechanics on board,—men whose interests lie in getting out work most expeditiously, rather than in doing work at a soldiering gait so as to avoid the trouble of beginning a new job, or of becoming fatigued even to a healthy degree. The point of view is essentially different.

At this period the aim should not be to set a high standard, but to pitch it so that the honest, efficient, faithful man may always succeed in accomplishing the task. Even this standard will show vast improvement in the rate of accomplishing work over the rate under our present gang system. By putting a

midshipman or other trained observer, with a stop watch, to timing a carefully-planned operation when a good mechanic is engaged on some one of the constant recurring overhaul or repair jobs, we will begin the collection of valuable data for our elementary time study sheets, from which we will soon be able to prescribe scientific standards determined experimentally.

STANDARD TIMES.

Because of the similarity of certain jobs on all ships, and because of the unity of purpose and practically equal ability and uniform training of all naval officers in charge of the various plants, all the more important time elements can be determined in a very short time. If, then, we classify the elements and adopt a uniform method of recording the data the concerted action of all the senior engineer officers will soon provide the data needed to make our standard times real standards of efficiency. Much data of this character already exists in many commercial plants. It can be obtained for the asking, and some of it may be applicable to our needs.

REDUCTION OF TIME INTERVALS.—GUNNERY PRACTICE.

It may be that objection will be offered to this minute subdivision and small times intervals; but we invite the attention of the objectors to our system of gunnery training and its results. In this training, as is well known to all officers of the service, the best thought and closest attention of more than 50 per cent. of all officers on board ship, as well as scores of the highest trained petty officers, are given to ascertaining and developing the best means of handling ammunition, loading, pointing and firing the guns. When these things are determined they are carefully written into detailed instructions. Every man is selected for a special place after careful study of the men available, then he is not only shown, time and time again, what to do, but just how and when to do it. The men being finally fitted, each into his specific place, days and weeks are devoted to perfecting him both as an entity and as a perfectly fitting cog in a smooth-running efficient gear train. During the training period every man's every movement is timed,

as well as every operation of the drill. The times taken are of the order of fifths of a second. To see a turret crew at drill before a record practice one could imagine that all the time keepers from a big athletic meet were present, so numerous are the stop watches.

It may safely be asserted that these phases of scientific management—scientific selection of men, careful planning, detailed instruction, and accurate time study—reach a higher development in the navy in gunnery training than they do in any private shop in the country. We all know the results. We all see and acknowledge the need for this infinite detail and painstaking instruction in matters pertaining to target practice; yet when it is proposed to employ these same methods in crude form for securing higher engineering efficiency, many of us doubt the necessity for going to so much trouble; or, admitting the need for increased efficiency, deny that methods which produce the highest gunnery efficiency will tend to engineering efficiency. Once it is fully realized that a ship's gunnery efficiency may and will be entirely nullified by lack of engineering efficiency then we will have not only the passive support but the active assistance of all officers in the service.

MILITARY EFFICIENCY THE ULTIMATE AIM.

Unfortunately, too many of us look at all repairs from the point of view of dollars and cents only. The only proper view point in the navy is *military efficiency*. We must have a full realization of the effect of a breakdown, even a small one, in the hour of battle; and must value each moment a ship is disabled due to breakdowns in terms of battle-moments. If this be the accepted value, then the question of preventing breakdowns, and repairing efficiently those which no foresight can prevent, have assumed their proper importance, and there will be none to question the advisability of going to any length to accomplish these vital ends. It is the province of the engineer to prevent breakdowns, that of the manager to repair them. Every officer charged with the responsible operation of the machinery plants of our heavy armored ships should be both an engineer and a manager.

INTERDEPENDENCE OF NAVY-YARD AND SHIP ORGANIZATIONS.

Accepting the dictum that the ship's force can and should keep the ship in a state of efficient preparedness during her entire commission, barring an accident to some major machine or part, it is still necessary to tie the ship and navy yard administration together in some manner to provide for such emergencies. The system of management herein outlined coupled with making the senior engineer officer an assistant, for duty on his own ship, to the yard management when his ship is at a navy yard, provides all that is necessary to insure the highest possible efficiency. The system insures that: All officers and responsible men on board have an intimate knowledge of the department, machinery, parts and men, together with their capabilities and needs. All work within the capacity of the ship's force will be done; all requests will be specific, and will be made only for work beyond the capacity of the ship's force. These requests will state specifically what is to be done, giving bills of material, and drawings or tracings showing changes. They will state what parts can be done by the ship's force.

ESTIMATES FOR NAVY-YARD WORK.

Due to the training incumbent under this system many officers on board will be qualified to submit careful and fair estimates of time and cost of work requested. These carefully prepared requests, accompanied by estimates, bills of material and drawings, may be forwarded to the home yard well in advance of the arrival of the ship. After being checked by the yard authorities they will be accurate enough to enable the Bureaus or Department to decide whether the work is to be authorized. They will also enable the yard authorities to make preliminary plans of the work and procure the necessary material.

SHIP'S OFFICERS TO RETAIN CHARGE WHILE AT NAVY YARDS.

During the entire progress of work the senior engineer officer will have charge of all work being done on board the ship either by yard men or by the ship's force. He will deter-

mine what is to be done by the ship's force towards facilitating the work on that part being done chiefly by the yard, always following the yard plan if there is a plan or if the work is covered by instruction card. He will be responsible for the employment of all men actually on the ship, for the workmanship, and for testing the machine or part, on the completion of the work.

Under this arrangement such indefinite and disgraceful requests as "Overhaul anchor engine;" "Overhaul steam launch machinery;" or "Overhaul boilers," will pass into history. There will be no division of responsibility. The officer who, by the regulations, must make the repairs and operate the machinery when the ship is away from the yard, is made absolutely responsible for installing all work when the ship is at the yard. His interests are paramount to any other, because he must make good defects that develop after the ship leaves the navy yard. This assignment is parallel with the Department's policy of sending officers who are to be heads of departments on a new ship to the ship-building yard for inspection duty on the ship on which they are to make their next cruise.

SAVING THROUGH CO-ORDINATION OF SHIP AND NAVY YARD.

The money saved by the above logical coördination of duty will be very considerable; but we must not lose sight of the fact that the chief value is a military one. We have placed the repair work on board ship directly under the ship's officers, who are best fitted to know the needs and meet the requirements. The responsibility that is inseparable from their duty when on the sea is continued in full force while the ship is of necessity having the same kind of work done with the assistance of the yard mechanic. The time the ship is disabled is greatly reduced. This, in effect, adds one ship to the fighting strength of the fleet for a period equal to the time difference between her disablement under present methods and her disablement under logical scientific management.

REPAIR PLANT OF THE U. S. S. *GEORGIA*.

By Lieutenant Commander Louis Shane, U. S. Navy, Member.

In these days of self-supporting ships it may not be amiss to describe some of the work done on one of them. The repair plant of the *Georgia* is fairly complete, and consists of a foundry, blacksmith's shop, copper shop, plumber's shop, carpenter's shop and machine shop.

FOUNDRY.

The foundry is situated on the upper deck and is in a small deck house abaft the after smoke pipe. It contains a Mircs combined blacksmith's forge and melting furnace capable of taking a No. 70 crucible, a bin with one-half a ton of Albany sand, several flasks, iron and wood, and other foundry accessories, such as tongs, shanks, clamps, etc. The largest casting made up to date weighed about 175 pounds with riser and gate. Since its original installation, in August, 1910, the foundry has turned out over 800 castings of all sizes. Among these are the castings for a speed lathe for the carpenter's shop, which include head and tail stocks, two tool posts, and two tool rests, a circular saw, cylinder heads for main circulating pump, blower engine and steam launch, cleats of all sizes, heads for oil strainers, distiller glands, eccentric straps for 100-k.w. dynamo engines, casting for arm of Blake pump, door for dynamo casing, crosshead for auxiliary condenser, 2-inch tees, brazing flanges for copper pipe, Lewis bolt-socket plugs, flanged cross for steamer, pistons for dynamos, and many others.

Oil fuel is used for melting metal. There is a connection for a portable oil burner, which was made on board. Albany

Fig. 1.—General View of Foundry.

sand is used, as practically none but brass castings are made. Iron castings would be made in emergencies, but the crucibles, when melting iron, last for only about three heats, while with brass they last for thirty heats. When at a navy yard advantage is taken of the facilities of the yard foundry to make iron castings. The ship's moulder does the work, and the

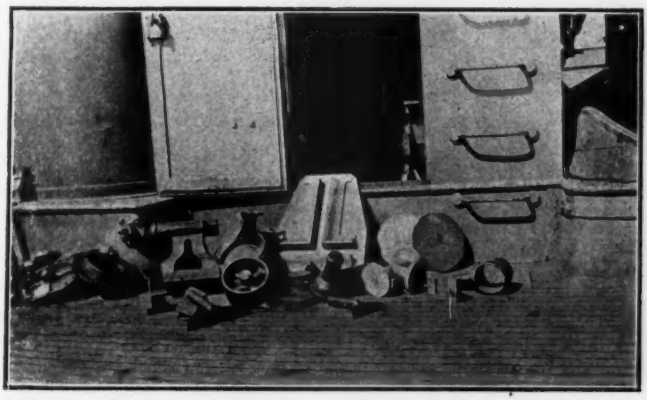


Fig. 2.—Patterns.

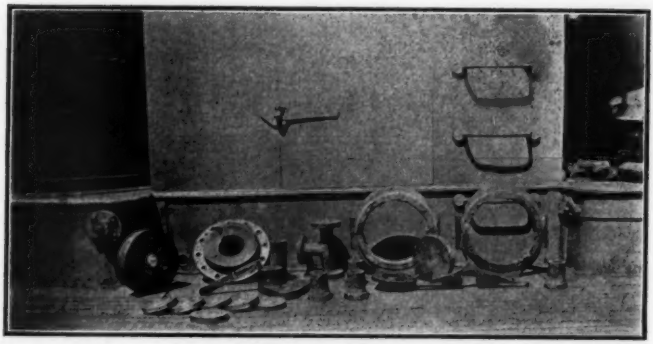


Fig. 3.—CASTINGS.

metal used is returned to the yard foundry in the form of pig stubbed out of store. Of iron castings so made the tuyeres for the large double forge in the blacksmith's shop, castings for a rigging jack for making wire splices, saddles for gap lathe, and dynamo pistons are examples.

Scrap metal is generally used to make castings. All brass, copper and other scrap metal is turned into the foundry by all departments. The red and yellow brasses are separated.

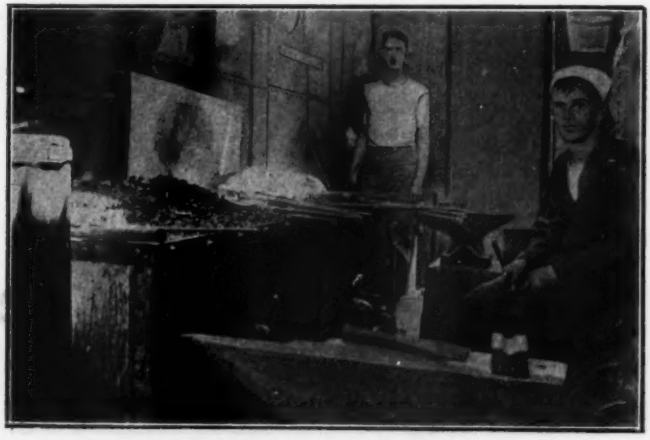


Fig. 4.—BLACKSMITH'S SHOP.

The old zincs discarded from boilers are melted and used for making bronze or brass mixtures of known constituents. Babbitt metal is generally used instead of tin for making 88–10–2 mixture, as there is no tin on board. The small amount of antimony does not seem to affect the metal adversely.

All patterns are made in the carpenter's shop, and the circular saw and speed lathe, built by the ship's force, have made pattern making very much easier than formerly. There are no rated pattern makers on board, but the wood workers of the carpenter's gang manage to turn out very creditable patterns. Altogether the foundry has been a great asset to the repair plant of the ship, and much work that formerly was done by the Navy Yard is now done on board, on account of the fact that castings can be made on board.

BLACKSMITH'S SHOP.

The blacksmith's shop is situated in a small deck house on the upper deck between the two forward funnels. Originally it had a small round forge with one anvil, and there was always contention as to whether the deck or engineer's blacksmiths should use the forge. There are three blacksmiths on board and there was only the one forge in the shop. There are portable forges on board, but at best these are mere makeshifts, the upper deck being the only deck capable of standing the heavy hammering necessary in blacksmith's work. The round forge was removed, and a large rectangular double forge built by the ship's force in its place. Each fire is larger and better than the old one, and 4½-inch bars are easily handled in them. A small ¼-H.P. portable electric blower furnishes the blast and is ample for both fires. All parts of the forge were made on board, including tanks, tuyeres, air pipes, etc. The fires can be used by the coppersmith for brazing flanges on 10-inch pipe. Such large brazing jobs are done on the blacksmith's forge, though smaller ones are done to better advantage with the portable burner. Both fires are

constantly at work, and there is now no controversy between the blacksmiths, as to who shall use the forge. As an example of the work done, an awning stanchion 3½ inches in diameter was made complete on board. Heavy bar iron was forged for the top and bottom and welded to a 3-inch pipe. The socket was cast in the foundry.

COPPER SHOP.

A small space in the trunk to forced-draft blowers is used as the copper shop. In it there is a portable bench which can be folded out of the way when the blowers are used for forced draft. All the small work is done here. The heavier work is done either in the foundry or the blacksmith's shop. The work is set up on the upper deck in a manner similar to that employed in building pipes to template on the floor of a copper shop on shore. The two coppersmiths on board have kept practically all the copper piping in repair, so that at present there is little to be done in copper work. The portable burner is of copper, made on board, and is used for all the smaller brazing jobs. Some of the work done by the coppersmiths is as follows:

On the trip to France the nozzle on main exhaust pipe connecting to bleeder pipe was found cracked throughout the circumference, seriously impairing the vacuum. On arrival at Brest the lagging was removed, the main exhaust pipe taken down on the floor, and a new nozzle put in place of the old one. This job is as good as could be done ashore and has given no trouble since.

The Navy Yard, Philadelphia, manufactured eight sanitary scuttle butts, one for each fireroom. None were made for the engine rooms, so these were made by the coppersmiths, exactly like those for the firerooms with one exception, viz: the Navy Yard used cork as lagging, and the ship's force used sawdust (obtained from the saw in the carpenter's shop).

Several branches of salt-water piping were renewed by the coppersmiths. When possible the old flanges were used, and

when it was not practicable to do so, new flanges were cast in the foundry.

The oil wells of the two main engines were cross-connected by the ship's force, the coppersmiths making up the necessary lengths of copper pipe. In several instances when pipe of the required size was not on hand the coppersmiths built it up from sheet metal. A cooling tank for cooling water before putting into the ice-freezing tanks was made by the coppersmiths. They also made forty-eight vaclite lamps of copper.

The foregoing are only a few of the more important jobs done by the coppersmiths. There are only two, and it is neeedless to say that they are kept busy.

CARPENTER'S SHOP.

This was originally in the small deck house now occupied by the foundry. This space was much too small, and the shop was transferred to a forward compartment on the port side of the gun deck. Large, heavy carpenter's benches were

Fig. 5.—CIRCULAR SAW AND SPEED LATHE.

installed on the outboard and after sides of the shop. Inboard are the speed lathe and circular saw. The lathe can swing a piece 17 inches by 8 feet. The saw can cut through 6-inch

oak with ease. A portable 3-H.P. electric motor is used to drive both machines. The addition of these two machines has more than trebled the capacity of the carpenter's shop.

PLUMBER'S SHOP.

The plumber's shop occupies the space on the starboard side opposite the carpenter's shop on the port side. The dish-washing machine tank was recently renewed by the plumbers. This and similar work is constantly being done by them.

THE MACHINE SHOP.

The machine shop is situated in probably the hottest part of the ship. It is directly underneath the evaporator room, just abaft the after firerooms, and just forward of the upper engine rooms. It has been considerably improved in habitability by the transfer of the four steam air compressors to the engine rooms.

There are the following tools:

1 large gap lathe capable of swinging 54 inches in the gap.
1 14-inch Flather machine lathe.
1 16-inch by 12-foot Hendey-Norton toolroom lathe.
1 No. 14½ Becker-Brainard universal milling machine with vertical attachment.
1 18-inch shaper.
1 28-inch drill press.
1 12-inch sensitive drill.

All these were in the original equipment with the exception of the Hendey lathe and the vertical attachment to the milling machine. A reamer grinder was in the original installation, but was removed for lack of room, and its place taken by a small electric center grinder. These machine tools, with one exception, are driven by one large shop motor through belts and counter shafts. The Hendey lathe has independent motor drive and is the most valuable tool in the shop. Another

similar lathe could be profitably used. All the tools are in excellent condition and in constant use. The machine-shop force consists of one chief machinist's mate, 2 machinist's mates, 1st class, three apprentices, and one coal passer as a general factotum. No other men are permitted to work in the machine shop. All work that requires machining is done by the machine-shop force for all stations and for all departments. The tools are in racks in lockers and easily accessible. Each tool has its place, and a glance will show whether it is in use or not. All these racks, lockers, etc., were installed by the machine-shop force. There is also a regular tool room

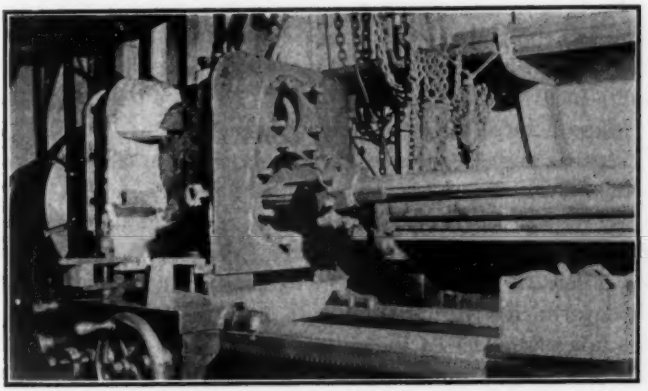


Fig. 6.—BORING 100-K.W. DYNAMO CYLINDER.

with its tool-room keeper, as in shops on shore. No tools are issued except on check, and once each week all tools are turned in and accounted for. This makes it easy to keep track of tools, survey and replace missing and broken ones, and place responsibility for those lost.

It is surprising what large work can be done in a well-equipped shop on board. Two cast-iron saddles were made by the ship's force for the large gap lathe. These secure by means of bolts and slots to the tool carriage. By using the boring bar the lathe can be made to serve as a horizontal boring mill. Three dynamo cylinders and the ice-machine compressor cylinder were rebored in this manner.

The following are examples of work done in machine shop:

Turned down starboard I.P. piston rod. (Had been scored by broken packing spring.)

Rebored several dynamo cylinders and ice-machine cylinders.

Made new ice-machine valve rock shaft complete.

Machined all parts of speed lathe and circular saw.

Machined rigging jack for making wire splices.

Manufactured range transmission device for sub-stations.

Made new range discs for turrets.

Took out lost motion in 12-inch turret-turning gear.

Took out lost motion and made new parts for 12-inch rammers.

Machined brackets for holding sub-caliber rifles on 12-inch and 6-inch guns.

Manufactured pistons for dynamo engines.

Manufactured eccentric straps for dynamo engines, and many other jobs too numerous to mention.

Work for the Engineering Department is done by verbal order. All work for other departments is done on regular job order. A card-index record is kept of all work done for all departments, including the Engineering Department, and this takes the place of the old work book.

OTHER WORK.

The rest of the work is done by the mechanics of the various stations. In the engine room the regular force completely retubed the main condensers in 105 working hours. They installed a great many additional splash plates since the installation of the forced-lubricating system, and cross connected the oiling systems. All pumps are being completely overhauled, and most of them will have their steam-valve seats refaced by the ship's mechanics. In the firerooms several boilers have already been relagged, and all will be completed during the repair period. All the brick work is being renewed, bunkers are being scaled and painted. All the ash pans and bilges are

scaled and painted quarterly, and the wisdom of this is shown by the fact that the original ash pans are still in the boilers after five years of commission, and will probably still be in use several years hence. The bilges are also in very good condition, there being comparatively little corrosion.

The *Georgia* has been assigned the first repair period, during July and August. The only work to be done in the Engineering Department by the Navy Yard is the manufacture of two castings for the ash-ejector system, and rebushing the stern bearings. All the other work is to be done by the ship's force, including relagging the remaining seventeen boilers, scaling and painting the bunkers, refacing steam-valve seats of fourteen feed pumps, and generally overhauling the department. There has been very little work done by the Navy Yard in the Engineering Department of the ship since the installation of the forced-lubricating system and the foundry, about a year ago. From the present outlook it will probably be a long time before any considerable work will be required that is beyond the capacity of the ship's force. Of course, this will depend on the Department's keeping the complement full, for, whatever the facilities in the form of tools, it takes men to do the work, a fact that is sometimes lost sight of.

In conclusion, the writer wishes to state that in his opinion the installation of the foundry and a few additional tools has been the prime factor in making the ship so nearly self supporting. Many castings are made on board, and, of course, are machined there. Before the installation of the foundry, if it were necessary to ask for a casting, the request would call for the finished product. Now, not only the cost of the casting, but also the cost of machining is saved. The foundry has certainly, on the *Georgia,* been the largest factor in reducing the cost of upkeep of the Engineering Department.

U. S. S. *UTAH.*

DESCRIPTION AND OFFICIAL TRIALS.

By Lieut. Commander William Russell White, U. S. N., Member.

The *Utah* (Battleship No. 31) built by the New York Shipbuilding Company, Camden, N. J., is one of two ships of this class authorized by Act of Congress May 13, 1908, her sister ship being the battleship *Florida,* building at the Navy Yard, New York.

The contract was signed November 24, 1908; the keel was laid February 22, 1909; hull launched December 23, 1909. The official trials occurred June 26 to July 2, 1911, and the vessel will be delivered to the Government about September 15, 1911, about two months over the contract time of thirty-four (34) months.

The contract price for the construction of hull and machinery, was $3,946,000, of which sum $2,846,000 was allotted to hull and $1,100,000 for the propelling machinery and auxiliaries. The contract required a speed of 20¾ knots to be maintained with a displacement of 21,825 tons. The allotment of weights of machinery was 2,060 tons. The actual weight was about 2,064.221 tons, or 0.379 ton below weight, the authorized changes adding 4.6 tons to the contract weight. The following trials are specified in the contract:

(*a*) A progressive trial over a measured-mile course, for standardizing the screws.

(*b*) A full-speed trial of four hours duration. The speed developed by the vessel upon this trial shall not be less than an average of 20¾ knots an hour, the air pressure shall not exceed an average of 2 inches of water, and the steam pressure

in the high-pressure turbine shall not exceed 185 pounds above atmosphere.

(*c*) An endurance and coal-consumption trial of twenty-four hours. The speed developed by the vessel upon this trial shall not be less than an average of 19 knots an hour. Careful record shall be made of the coal consumed. All necessary auxiliaries shall be in operation, including all those usually required under service cruising conditions, and especially such dynamos as may be necessary for efficiently lighting the vessel and operating at their normal speed the hull ventilating fans. The evaporating and distilling plant shall be in continuous operation at its normal rated capacity. The pumping and forced-draft air system will also be operated as required.

(*d*) An endurance and coal-consumption trial of twenty-four hours, at an average uniform speed of 12 knots. The trial shall be made as nearly as possible under cruising conditions, and the coal consumption for all purposes carefully determined.

The water consumption on trial (*b*) shall not exceed 15.0 pounds; on trial (*c*) 16.25 pounds, and on trial (*d*) 26.0 pounds of water per hour per horsepower, including shaft horsepower of turbines and indicated horsepower of all engineer's auxiliaries in use on the trials, except evaporating and distilling plant.

During all the official trials all the exhaust steam and all the drains except from the evaporators passed through the fresh-water measuring tanks.

HULL DIMENSIONS.

Length between perpendiculars, feet and inches	510–00
on L.W.L., feet and inches	510–00
over all, feet and inches	521–06
on straight keel, feet and inches	460–00
Projection forward of F.P., feet and inches	11–06
aft of A.P., feet and inches	0–00
Breadth, extreme, at L.W.L., outside of armor, ft. and ins.	88–02½
molded, feet and inches	87–10½
Depth, molded, main deck at side M.S., feet and inches	44–05½

Ratio of length to beam	5.781
Displacement per inch at L.W.L. tons of S.W., tons	73.82
Area of midship section, square feet	2,458
L.W.L. plane, square feet	31,056
wetted surface, square feet	56,700
Coefficient of fineness, block	.5837
Length of fireroom space, feet and inches	102–00
engine-room space, feet and inches	60–00

MAIN ENGINES.

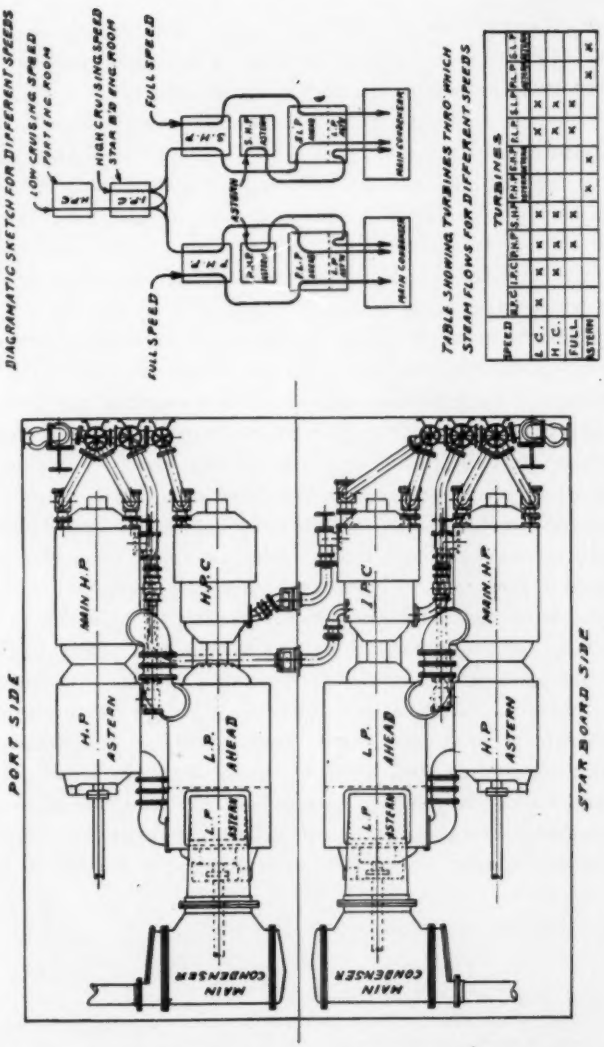
The propelling machinery consists of Parsons turbines arranged on four shafts, six turbines being provided for going ahead and four for going astern, the disposition of the turbines being as follows: On the starboard and port outboard shafts are the main H.P. ahead and the H.P. astern; on the port inboard shaft is the H.P. cruising turbine, one L.P. ahead and one L.P. astern; and on the starboard inboard shaft is the I.P. cruising, one L.P. ahead and one L.P. astern. The main H.P. ahead and cruising turbines are placed forward of the H.P. astern and L.P. turbines on their respective shafts. The L.P. ahead and L.P. astern turbines are contained in the same casing.

The turbines mentioned are arranged for use in the following combinations:

Full Speed Ahead.—Steam is admitted to the two H.P. ahead turbines, from which it passes to the two L.P. ahead turbines, exhausting to the main condensers. Under this condition all of the cruising and astern turbines revolve in a vacuum, this vacuum being maintained through the large drain connection on these turbines.

High Cruising Speeds.—Steam is admitted to the I.P.C. turbine, from which it passes into the two H.P. ahead, then to the two L.P. ahead turbines, exhausting to the main condensers. Under this condition, steam not being admitted to the H.P.C. or the astern turbines, they revolve in a vacuum, which is maintained through the large drain connections, the connection between the H.P.C. and I.P.C. turbines being closed by a self-closing valve.

Low Cruising Speeds.—Steam is admitted to the H.P.C.



U. S. S. UTAH.
Plate 1.

turbine, from which it passes to the I.P.C. turbine, thence into the two H.P. ahead turbines, thence into the two L.P. ahead turbines, exhausting to the main condensers.

Going Astern.—Steam is admitted to the H.P. astern, thence to the L.P. astern turbine, exhausting to the main condenser, each engine room being independent of the other.

By-pass valves are provided for passing steam around the first expansion of the main H.P. turbines, but these were not used on the trial and will not be necessary at any time. They should be removed, being cumbersome and unsightly.

The exhaust pipes between the H.P.C. and I.P.C. and also the I.P.C. and both H.P. ahead turbines, are provided with screw-down self-closing valves.

The general arrangement of turbines and the flow of steam when using different combinations are shown on Plate I.

Turbines.—Each turbine consists of two essential parts, the first, or fixed part, being the cylinder, or casing, and the second, or moving part, the rotor, supported at each end by a bearing connected to the lower half of the cylinder. In the cylinder are located the guide blades and in the rotor the moving blades. The cylinders are of hard, close-grained cast iron, divided into two parts at the axis on a horizontal plane, the lower half being provided with feet for bolting to the seating. The method of machining the cylinders was to rough bore and plane, after which the parts were bolted together and given the maximum hydrostatic test. The castings were then heated by steam at atmospheric pressure for twenty-four hours to anneal them and then allowed to cool, after which the finishing work was done and the circumferential grooves cut for the guide blades. No hydrostatic tests were made after the cylinders were finished and bladed. The low-pressure casing is shown in Plate II.

The following hydrostatic tests were made:

Cylinder:	Pressure, lbs.
Main H.P. ahead and H.P. astern	300
H.P.C. and I.P.C	300
L.P. ahead (steam and exhaust ends) and L.P. astern	30

Plate III shows one of the L.P. rotors.

Each rotor consists of a drum of class B forged steel, securely fastened at each end to a wheel, or spider of class B cast

Plate II.—Low-Pressure Turbine Casing.

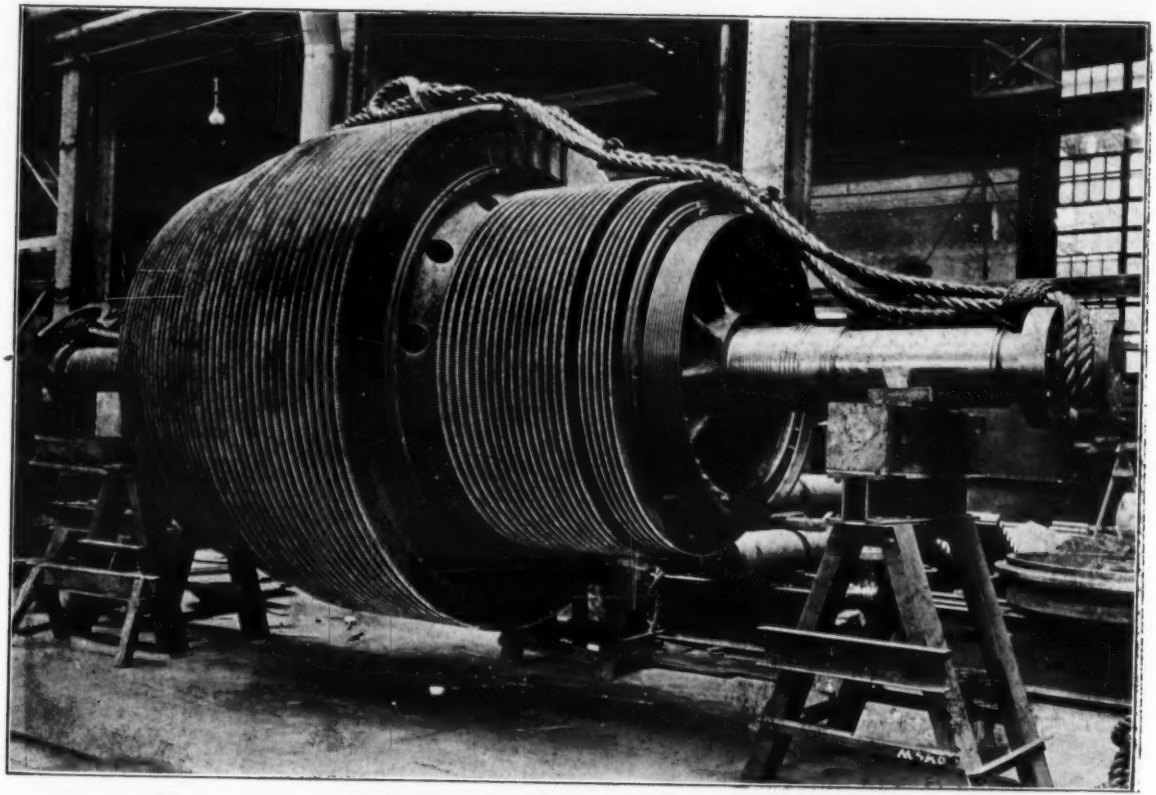


Plate III.—Low-Pressure Turbine Rotor.

steel, that is shrunk on and keyed to the rotor shaft made of class A forged steel, the forward section of shaft for each main H.P., H.P.C., I.P.C. and L.P. ahead turbines being provided with thrust collars at the forward end to take the propeller thrust. On the external surface of the drums are circumferential grooves into which are fitted the moving blades. Great care was taken to insure the rotors being perfectly balanced when the work was entirely completed, first, statically on knife edges and, second, dynamically under steam, using its own casing at the maximum working revolutions, the turbines being located on a surface plate in the machine shop. There was found to be such a minimum of vibration as to make it possible to balance upright a five-cent piece on the casing flange.

The blading is in accordance with the Parsons system, the blades and binding strips being 72 per cent. copper and 28 per cent. zinc, the calking strips 63½ per cent. copper and 36½ per cent. zinc, and the lacing wire of copper. The binding strips are fastened to the blading with silver solder, the blades, during the process of calking, being carefully straightened and the ends chamfered off and turned to the required diameter to give the necessary clearance for the various stages of expansion.

The dummy and rotor-shaft glands are steam packed, the dummy packing consisting of brass rings with a hook-shaped knife edge having a clearance of about .02 of an inch from grooves turned in the rotor sleeve; and the gland packing consisting of straight knife-edged packing rings inserted alternately in the gland sleeve and rotor, the clearance being about .015 of an inch.

A micrometer for measuring the dummy clearances is provided at the forward end of each ahead turbine.

Main Bearings.—There is one main bearing at each end of each turbine. It consists of a pedestal bolted to the turbine casing, having a bottom brass that can be rolled out without lifting the shaft and a top brass that will lift out with the cap. The brasses are of composition G lined with babbit metal, the lubrication being under pressure as described elsewhere in this article.

Plate III.—Low-Pressure Turbine Rotor.

Thrust Bearings.—Thrust bearings are provided as referred to above, being carried by the same pedestals as carry the main bearings. They consist of semicircular bushings of class B forged steel, in halves, the lower half taking the forward thrust and top half the astern thrust. The thrust rings consist of semicircular rings set into rings in the bushings, the Parsons Co. mixture for such purposes being used, consisting of copper, 6 parts, tin 1 part, and lead, ¼ part. The lubrication is as noted for the main bearings.

A governor is provided at the forward end of each shaft. The governor is of the Proell type, adusted to operate at about 400 revolutions per minute, the leads from the governor in each engine room uniting in one common lead to the main engine-room stop valve, located at the main steam separator at the forward outboard end of the engine room. The stop valve is of the Willits balance type, that has been described in this JOURNAL, and is provided with two small micrometer valves on the balance-piston side of the stop valve; one valve controlling the steam pressure to the under side of the balanced piston and the other connecting this space to the condenser. When the governor gear trips, the micrometer steam valve is closed and the valve connections to the condenser opened, thus closing that main valve instantly and shutting off all steam to the main engines.

Turning Gear.—An electrically-operated turning gear is provided for each shaft. It consists of a 5-H.P., 120-volt, 300 to 600-r.p.m. reversible Diehl Mfg. Co. motor, connected by a pinion and spur gear to right and left-hand miter gear wheels engaging a miter gear on a shaft that carries a worm engaging the worm wheel on the main shaft. The worm wheel is carried by feathers on its shaft and can be dropped out of mesh when the turning gear is not in use. Provision is also made for turning by hand by means of a ratchet. The controller is located close to the turning gear, and consists of a drum cylinder, rotating under copper fingers, a main switch mechanically interlocked with the operating shaft of the controller, and a field resistance contact board. There is but one operating

handle, and all functions are obtained on the controller by a movement of its handle.

Turbine-Lifting Gear.—The lifting gear consists of two 20-ton Sellers Co. trolleys on a track under the protective deck over each shaft, running the full length of the turbines. The mechanism that operates the lifting blocks is applied by hand through the medium of chains of sufficient length to reach the engine-room floor. In order to lift the upper half of the cylinders and the rotors squarely guide rods are used, these being located in the bottom half of the cylinder casings and engaging brackets in the top half. The guide rods are graduated so that both ends are lifted evenly. When the upper half of the cylinder is lifted clear of this rotor it can be suspended by rods fastened to the trolley-track structure and located directly over the guide rods This done, this rotor can be lifted clear of the lower cylinder casing and suspended by the rods above referred to at sufficient height to enable a complete examination of the blading or for reblading, if this is necessary.

MAIN TURBINE DATA.

	Diameter.	Length
Motor drums:		
Main H.P., inches	71	114
H.P. cruising, inches	71	63½
I.P. cruising, inches	70	72½
L.P. ahead, inches	97	87⅝
H.P. astern, inches	71	36½
L.P. astern, inches	71	44⅛
Number of expansions:		
Main H.P. and L.P. ahead, each		6
H.P.C. and I.P.C., each		3
H.P. astern and L.P. astern, each		4
Turbine casings, diameter, inches, each expansion:		
Main H.P.	73½, 74½, 75½, 77½, 80, 83½	
L.P. ahead	107½, 111½, 117½, 128, 128, 128	
H.P.C.	72½, 72⅛, 73½	
I.P.C.	72½, 73½, 74½	
H.P. astern	72⅛, 73⅛, 75, 77	
L.P. astern	80, 83½, 83½, 83½	
Length of casing for each expansion and diameter noted above:		
Main H.P., inches	17½, 17, 17⅛, 18⅛, 19½, 24 9/16	
L.P. ahead, inches	12⅛, 13, 15 3/16, 19, 19, 19	
H.P.C., inches	18½, 18½, 25½	
I.P.C., inches	23½, 23½, 24⅛	

Length of casing for each expansion and diameter noted above :

H.P. astern, inches 9½, 8¾, 8¾, 9

L.P. astern, inches ..... 9¾, 11$\frac{23}{32}$, 11$\frac{23}{32}$, 11$\frac{23}{32}$

Rows of blading for each expansion :

Main H.P 13

L.P. ahead ..... 3 of 7, 3 of 6

H.P.C 20

I.P.C 18

H.P. astern 6

L.P. astern 5

Length of blades for each expansion, inches :

Main H.P 1⅛, 1⅜, 2½, 3½, 4½, 6½

L.P. ahead 5½, 7½, 10½, 13½, 13½, 13½

H.P.C ..... ⅝, 1$\frac{3}{16}$, 1$\frac{7}{16}$

I.P.C 1½, 1⅝, 2½

H.P. astern ..... ⅞, 1⅝, 2, 3

L.P. astern 4½, 6½, 6½, 6½

Rotor shaft and bearings:	Length, ins.	Diameter, ins.	Diam. axial, ins.	Length over all of rotor drum and shaft, ft. and ins.
Main H.P.	18½	14	9	21–07½
L.P. ahead	29	15	11	25–10½
H.P.C.	14½	14	11	15–02½
I.P.C.	14½	14	11	16
H.P. astern	12½	14	9	11–02½
L.P. astern	29	15	11	With ahead.

Thrust bearings :	M.H.P. each.	I.P., one.	H.P.C.	I.P.C.
Collars on shaft, number	17	17	8	8
Thickness, inches	00½	00½	00½	00½
Distance between, inches	01$\frac{1}{16}$	01$\frac{1}{16}$	01$\frac{1}{16}$	01$\frac{1}{16}$
Outside diameter, inches	17½	17½	17½	17½
Inside diameter, inches	12½	12½	12½	12½
Bearing surface, top, square inches	803.7	803.7	351.6	351.6
Number of shoes, top	16	16	7	7
bottom	17	17	8	8

Effective thrust surface (oil grooves out) :

Full speed ahead, square inches 3,415.87

High cruising speed, square inches 3,817.67

Low cruising speed, square inches 4,219.47

Number and diameter of steam and exhaust connections, each turbine :

	M.H.P.	H.P.C.	I.P.C.	H.P. astern.	L.P. ahead	L.P. astern.
Steam, ins.	2—10½	1—10½	2—10½	1—13	43	29
Exhaust, ins.	1—43	1—17	2—17	1—29	1—(48 square feet.)	

SHAFTING.

The shafting of the outboard shafts is made in three sections and that of the inboard shafts in four sections, all of Class A forged steel.

The outboard coupling consists of a sleeve, in halves, secured to the shaft by two keys, the halves being held together by bolts.

Data for Shafting.

Line shaft, diameter outside, inches	12½
axial hole, inches	08
journal, inches	12½
Stern-tube shaft, diameter outside, inches	12½
axial hole, inches	07½
Propeller shaft, diameter outside, inches	12½
axial hole, inches	07½
Strut bearing, diameter, inches	14½
length, inches	60
Stern-tube bearing, diameter, inches	14½
length, inches	60

Each propeller shaft is supported by one strut fitted with lignum vitae.

PROPELLERS.

The four propellers are each the same size and are solid castings of manganese-bronze, with three blades. They all turn outboard for the ahead motion.

The driving faces of the blades were machined to a true-screw surface by means of a propeller-planing machine and afterwards polished. The backing faces were ground smooth and then polished. Each propeller was run on a mandrel for proper balance, and any excess metal was removed from the backing faces.

The hub is taper bored and keyed to the shaft, and at the forward end there is fitted a gland with rubber packing. The nut is securely locked and is covered by a composition cap which is turned and polished outside. The clearance spaces are filled with tallow and all recesses on the outside are carefully covered with plates.

Data for One Propeller.

Diameter, feet and inches	9–02
Pitch, feet and inches	8–06½
Ratio of pitch to diameter	0.9284

Developed area, starboard, square feet	41.1458
port, square feet	41.041
Projected area, starboard, square feet	36.8333
port, square feet	36.77
Disc area, starboard, square feet	65.995
port, square feet	65.995
Developed area divided by disc area	0.62267
Height of lower tip of blade above keel, outboard, feet and inches	3–06½
inboard, feet and inches	3–03
Immersion of upper tip of blade (draught 30 feet aft), outboard, feet and inches	17–04
Immersion of upper tip of blade (draught 30 feet aft), inboard, feet and inches	17–07

TORSION METERS.

A Föttinger torsion meter is installed on each shaft for determining the shaft horsepower of the main turbines by measuring the torsional deflection of the shaft while in motion. Fig. IV is a sectional drawing of the working parts. The ring B with two arms and the ring D at the after end of the steel sleeve C are both clamped firmly to the shaft. On the forward end of the sleeve is a ring A similar to B. There is a clearance between this ring and the shaft, and it is centered on B by means of radial centering rods having knife edges on each end. The sleeve C and ring A act as a rigid mass and turn with that part of the shaft to which D is clamped. When a torsional stress is applied to the shaft the ring B twists relative to A. Through a flexible link E a bell crank H pivoted on A, and a link J connected to the magnalium ring N which floats on the sleeve C, this relative twist is magnified and changed to a fore-and-aft motion. A shoe P is brought into contact with a flange on the magnalium ring, and the fore-and-aft motion of the ring is again magnified and indicated by a pointer Q on a scale T.

The constant for the machine is found by calibrating the shaft and torsion meter in the shop previous to installation in the ship. The after end of the shaft is bolted to a fixed flange. The forward end rests in a roller bearing and on the forward flange is bolted a double lever. A known twisting moment is

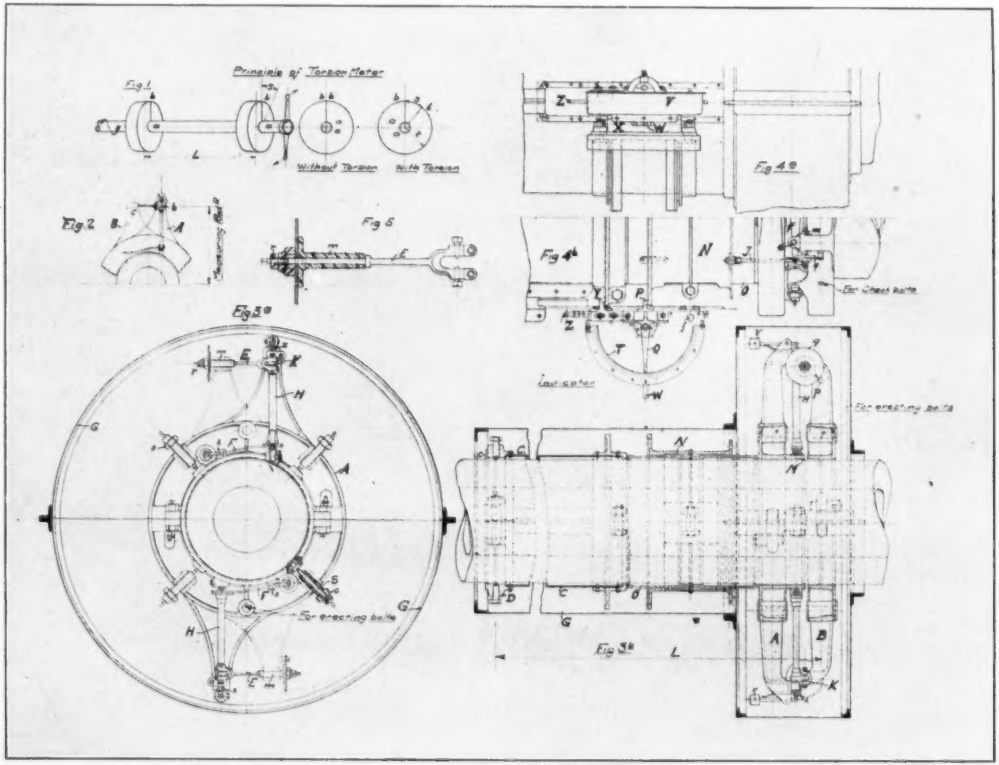


Plate IV.—DIAGRAM. FÖTTINGER TORSION METER.

applied to this lever and the corresponding torsion-meter readings are observed. From the slope of the line plotted between torsion-meter readings and torque a constant C is obtained such that H.P.$=C\times R\times F$, in which H.P. is shaft horsepower, R is the number of revolutions of the shaft per minute, and F is the torsion-meter reading.

Shaft Calibration Data.

	1	*2*	*3*	*4*
Length calibrated, feet and inches,	27–00	23–02½	23–02½	27–00
Foot pounds for one inch torque on 120-inch radius	44,137.93	51,405.62	51,405.62	44,599.3
Length of shaft which torsion meter records, inches		68.112		
Radius of arc on which torque is read, inches		14.96		

STEERING ENGINE AND GEAR.

The steering-engine room is located on the starboard side of the center line, just abaft frame 95, access being from the starboard engine room. The engine is vertical double, with 17 by 14-inch cylinders, constructed by the Hyde Windlass Company, Bath, Me., and is controlled from the steering stands in the conning tower, on the upper conning platform and in the central station through wire-rope transmission, and also by telemotor.

The engine may also be controlled by a hand wheel in steering-engine room and from the hand-steering room. The various controls are connected by clutches, so that any one may be operated independently of the others. The steering-engine shaft is continued aft to the hand-steering room at frame 115, where are located four large wheels for hand steering, provided with suitable clutch, so the handwheels may be disconnected when not in use.

The tiller room is immediately abaft the hand-steering room, the main steering gear consists of a right-and-left-handed screw on which are two driving nuts direct connected by side

rods to the crossheads on the rudder stock. The screw is operated through gearing by either the steering engine or the handwheels, either of which may be disconnected by clutches.

WATER SERVICE.

The water service consists of one 1½-inch connection from the discharge side of the main circulating pump and a 1½-inch connection from the fire main.

There are connections to the stern tubes and the main air pumps and to no other machinery.

A FORCED-LUBRICATION SYSTEM

Besides the regular oil-lubricating system a forced-lubrication system is provided in each engine room, the essential features of which are as follows:

Two piston oil pumps;

Two Schutte & Koerting corrugated-film oil coolers, shown on Plate V;

One Blake pump for circulating water in oil coolers;

One Macomb oil strainer in each oil-pump suction;

One 495-gallon drain and cooling tank.

The pumps take oil through the strainer from the drain tank and deliver it to the oil coolers just under the protective deck, a relief valve maintaining a pressure of about 20 pounds in the system. From the oil coolers the oil goes to the main-turbine line bearings, thrust and circulating pump engines and drains back by gravity to the drain tank.

Forced lubrication is applied to the top and the bottom of the main-turbine bearings and to both the forward and the after faces of the thrusts, and also to the circulating-pump engines. Oil from the main bearings is caught in oil wipers and that from the thrusts collects in a reservoir in the thrust pedestals, so that the bottoms of the collars run continuously immersed. In the drains from the oil wipers and in the overflow from the thrust-pedestal reservoirs are sight glasses

through which the flow of oil may be observed. Thermometers are fitted to these sight glasses to show the temperature of the oil leaving the bearings.

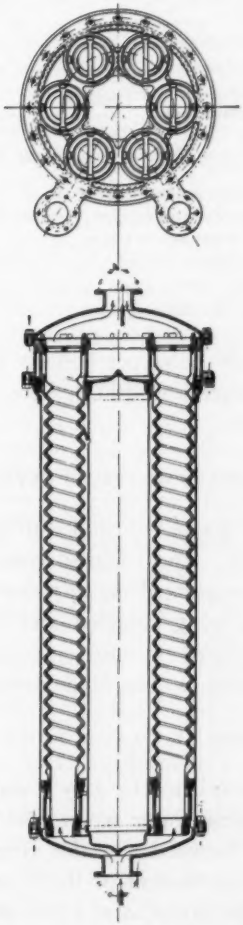


Plate V.—LUBRICATING-OIL COOLER.

Data for one Oil Cooler.

Weight, empty, pounds	1,335
Number of tubes	6
Outside diameter of inner tubes, inches	0$4\frac{7}{8}$
outer tubes, inches	0$5\frac{1}{2}$
Thickness of tubes, B.W.G	14
Length of tubes between tube sheets, inches	54
Thickness of tube heads, inch	0$0\frac{5}{16}$
Cooling-surface tubes, square feet	76
Total cooling surface, square feet	82

Data from Official Four-Hour Trial.

	Starboard.	Port.
Temperature of oil entering coolers, degrees F.	103.8	105.4
discharge oil from coolers, degrees F.	94.2	93.6
Double strokes of forced-lubrication pump	24.7	25
Pressure of oil entering coolers, pounds	156.5	177.8
discharging from coolers, pounds	12.9	12
Double strokes oil-cooler circulating pump	68.8	67

Capacity of one oil cooler per hour is 4,000 gallons of oil, cooling from 125 degrees to 100 degrees F., with circulating water at 70 degrees F.

ELECTRICAL FIRING DEVICE.

An electrical time device is installed to aid in the systematic firing of the boilers. The motor-driven transmitter in the engine room can be regulated for any desired period of firing the boilers, a gong being sounded and numbers flashed on suitable indicators placed on each side of each fire room to show when each furnace is to be fired or worked.

PIPE INSULATOR PUMP.

In each engine room is a 6 by 8 by 8 vertical, double-acting, single pump for forcing water around the flanges of the main steam pipes at the forward and the after bulkheads of the pipe passages. The discharges of these pumps are cross connected in the engine rooms, and after service deliver overboard on the port side of the after fireroom.

The engine-room fire and bilge pumps can also be used for pipe insulator purposes.

COUNTERS AND GAGES.

In each engine room is fitted two continuous-counter devices, one from each main turbine shaft, and two revolution indicators (tell-tales) having two pointers. The inboard tell-tale indicates the relative speeds of the inboard shafts and outboard tell-tales the relative speeds of the outboard shafts.

Dial steam gages are supplied for measuring the main steam line, all the turbines individually, vacuum, auxiliary steam, pressure on the feed-water lines, boilers, etc. A recording gage is connected to the main steam line.

AUXILIARY CONDENSER.

There is in each engine room one Blake horizontal combined air and circulating pump with a surface condenser.

Data.

Diameter of steam cylinder, inches	06
air cylinder, inches	10
water cylinder, inches	10
Length of stroke, inches	12
Diameter of circulating-pump suction, inches	06
discharge overboard, inches	06
air-pump, suction, inches	05
discharge, inches	04½
Cooling surface of each condenser, square feet	725
Number of tubes	661
Outside diameter of tubes, inch	00⅝
Length of tubes, inches	82½
Thickness of tubes, B.W.G.	16

MAIN CONDENSER.

There is one main condenser in each engine room, located at the after end, and the two are cross connected. They are 10 feet 0 inches in diameter by 14 feet 6 inches long, being 11 feet between tube sheets. The shell plating is of Class B boiler steel, and the water chests of composition G. Tubes of composition: 70 copper, 29 zinc and 1 tin. Tube sheets and tube-supporting plates of rolled naval brass. The sup-

porting plates are bushed with material of same composition as the tubes.

The outboard water chest, for the entrance and exit of the circulating water has a movable division plate on the horizontal center line of the condenser. The inlet and the outlet nozzles are 30 inches in diameter, with the inlet on the bottom.

Data for one Condenser.

Number of tubes	8,466
Thickness of tubes, B.W.G	16
Outside diameter of tubes, inch	00⅝
Length of tubes, between tube sheets, feet and inches	11–02½
Number of supporting plates	2
Cooling surface on outside of tubes, square feet	15,235
Ratio of cooling surface to total heating surface	.237

Data from Four-Hour Trial.

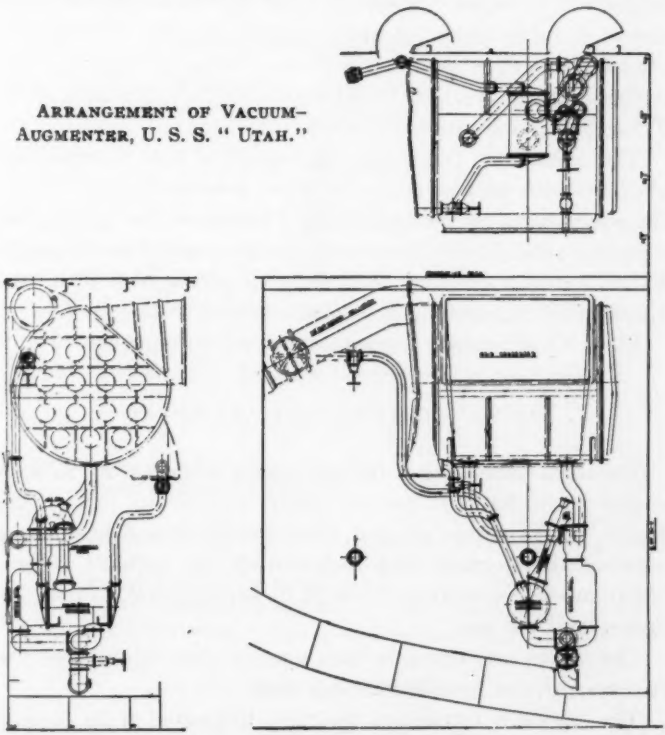
	Starboard.	Port.
Temperature of injection water, degrees F	68.7	69.2
discharge water, degrees F	93.3	92.8
Ratio of cooling surface to S.H.P. of main engines	1.127	

VACUUM AUGMENTER SYSTEM.

The vacuum augmenter, an adjunct of the main condensing system, is located at the after inboard end of each engine room, between the condenser and air pump. It consists of a jet box, a swiveling chamber, a water seal and a condenser, with the latter located under the inboard line shafting. The various parts noted are located in relation to the main condenser and air pump in the order named. Connection to the swiveling chamber of each main condenser is made by two 9-inch copper pipes, one of which on this vessel, contrary to the Parson Co.'s usual practice, is not carried inside the bottom of the shell so as to penetrate the steam space above the maximum water level. This means that the same amount of water can pass through both connections, and necessitates perfect functioning of the swiveling chamber and jet box to extract the vapor and drive it into the augmenter condenser. This arrangement would not have been used but for the fact that the condensers were completed before the use of an augmenter was decided

upon, and the necessary internal changes could not be made in accordance with established practice. As a vacuum of 29 inches was easily maintained with about 24 double strokes of the air pump the indications are that the design is satisfactory; but it would be interesting by way of a test to cut out the augmenter system for a series of trials under various steaming conditions, and drive the main circulating pumps harder.

The operation of the system is as follows: The two main condenser connections enter a light composition chamber, 16 inches inside diameter, located vertically, from opposite sides, the object being to so agitate the water by the two volumes



ARRANGEMENT OF VACUUM-AUGMENTER, U. S. S. " UTAH."

Plate VI.

coming in contact with each other as to cause the vapor to be extracted and rise to the jet box, consisting of a light composition, 180 degrees bend, 16 inches in diameter, located horizontally, forming an air chamber, and containing a small steam nozzle so located as to direct the flow of vapor to the augmenter condenser, while the water passes down through the swiveling chamber to the water seal or trap, consisting of a light composition casting containing two diaphrams that overlap each other in such a manner as to form two chambers, from the bottom of which connection is made to the air-pump suction, to which a connection is also made from the bottom of the augmenter condenser. Cooling water for the augmenter condenser is taken from the main circulating-pump discharge to the main condenser, and the overboard discharge is through the main-condenser overboard discharge inboard of the main valve. These connections are both 9-inch.

The augmenter condenser consists of a light composition shell 29 inches inside diameter by 4 feet between the tube sheets, the connections on the shell being 13-inch on the top for the cone from the jet box, 12-inch on the bottom to the air pump. Each condenser contains 685 tubes, ⅝-inch outside diameter, No. 16 B.W.G., totaling 450 square feet of cooling surface.

Plate VI shows the augmenter and its arrangement.

MAIN CIRCULATING PUMP AND ENGINE.

The main circulating pump and engine is located aft in each engine room, between the two shafts and on the lower platform. These pumps are not cross connected and have three suctions; main drain, 15½ inches; two sea, each 21 inches. There are two discharges, each 30 inches diameter, to the condenser and the sea.

The valves are locked in such manner that only one of the two systems can be operated at a time.

The engine is compound, and it is lubricated from forced-lubrication system, and is entirely enclosed from two inches

below the bottom cylinder head. There is a stuffing box through the casing, through which the elongated piston rod works.

Data for one Pump and Engine.

Capacity of pump, gallons per minute	15,000
Diameter of suction nozzle, inches	30
discharge nozzle, inches	30
impeller, inches	51
H.P. cylinder, inches	11½
L.P. cylinder, inches	23
Stroke, inches	12
Diameter of piston rods, inches	02½
Length of connecting rods, inches	12

Data from Official Four-Hour Trial.

	Starboard.	Port.
Average revolutions per minute	231	210
I.H.P. per minute	225	190

MAIN AIR PUMPS.

The main air pumps are twin, vertical-beam, bucket, single-acting, of the Blake design. They are located at the inboard after-end lowest point in each engine room, between the main condenser and the feed and filter tanks. The main air pumps are cross connected through the channel ways.

The discharge is into the filter portion of the feed tank.

Data for one Air Pump.

Diameter of steam cylinder, each, inches	17½
water cylinder, each, inches	35
Stroke, inches	21
Diameter of steam piston rods, inches	02½
air piston rods, inches	02½
suction from turbine drains, inches	05
nozzle drain, inches	12
discharge, inches	10

Data from Official Four-Hour Trial.

	Starboard.	Port.
Average (double) stroke, per minute	23.6	44.4
vacuum	28.09	28.2
I.H.P	14	25

FEED AND FILTER TANKS.

The feed and filter tanks are located in the inboard forward upper and highest corner of each engine room. The top is arranged as a filter, the filter material being "loofa," which is placed between perforated plates. The filtering chamber is divided by vertical division plates so arranged that the water flows under and over in succession. All material is steel, well galvanized before assembling.

A 7-inch overflow pipe leads from the bottom of the tank up through the top, and to the bilge through a funnel, so that any water passing down may be seen. On the outside of the tank is fitted a gage glass and a graduated brass scale for showing the height of the water.

Capacity of each feed and filter tank, gallons	4,700
Total capacity of the feed and filter tanks, gallons	9,400

MAIN FEED PUMP AND FEED SYSTEM.

There are two Blake vertical, double-acting, single, main feed pumps in each engine room, located inboard, on forward engine-room bulkhead, below the operating platform and at the lowest forward point. They have a head of about 8 feet from the feed tanks. The suctions are from the feed tanks, air-pump channels and fresh-water double bottom.

The discharge from each pair of main feed pumps unites in a 5-inch pipe, which later joins in the firerooms to the discharge from the other engine room into a 7-inch feed main leading to the forward boilers, diminishing in size as it goes forward in the forward fireroom to $3\frac{1}{2}$ inches.

The branch, main and auxiliary feed pipes to boilers are each $2\frac{1}{2}$ inches in diameter.

On each feed pump is a grease extractor which may be by-passed.

The discharge from each feed pump passes through a Schutte-Koerting corrugated-film feed-water heater, which may be by-passed.

Data for one Pump.

Diameter of steam cylinder, inches	14½
water cylinder, inches	09½
Stroke, inches	18
Diameter of piston and plunger rods, inches	02½
suction nozzle, inches	05
discharge nozzle, inches	04
Material of water end	Composition.

Data from Official Four-Hour Trial.

	Starboard.	Port.
Average double strokes per minute (2 pumps)	26.5	20.5
I.H.P. per minute	104	77
pressure on feed main, pounds	260	260

FEED-WATER HEATERS.

In each engine room on the outside bulkhead there is a feed-water heater of the Schutte-Koerting corrugated-film type installed horizontally on the discharge side of the main feed pumps. The materials of which these feed-water heaters are made is as follows: Tube headers, composition; shell, steel; water chests, composition; tubes, copper.

On Plate VII is a plan of these Schutte-Koerting feed-water heaters.

Designed capacity of one feed-water heater per hour 250,000 pounds of water from 90 to 200 degrees F. with steam at 10 pounds pressure.

Data for one Heater.

Weight, empty, pounds	6,800
Total heating surface, on outsides of tubes, square feet	258
Number of tubes	16
Outside diameter of inner tube, inches	05
outer tube, inches	05 23/64
Length of tubes between tube sheets, feet and inches	05–06
Thickness of tubes, B.W.G., outer	09
inner	10
Thickness of tube heads, inch	00 7/16

Data from Official Four-Hour Trial.

	Starboard.	Port.
Temperature of feed water, degrees F.	163	184
Average pressure in feed line, pounds	220	220
double strokes of feed pumps	26.5	20.5
Exhaust steam pressure in heaters, pounds	4.28	5.75

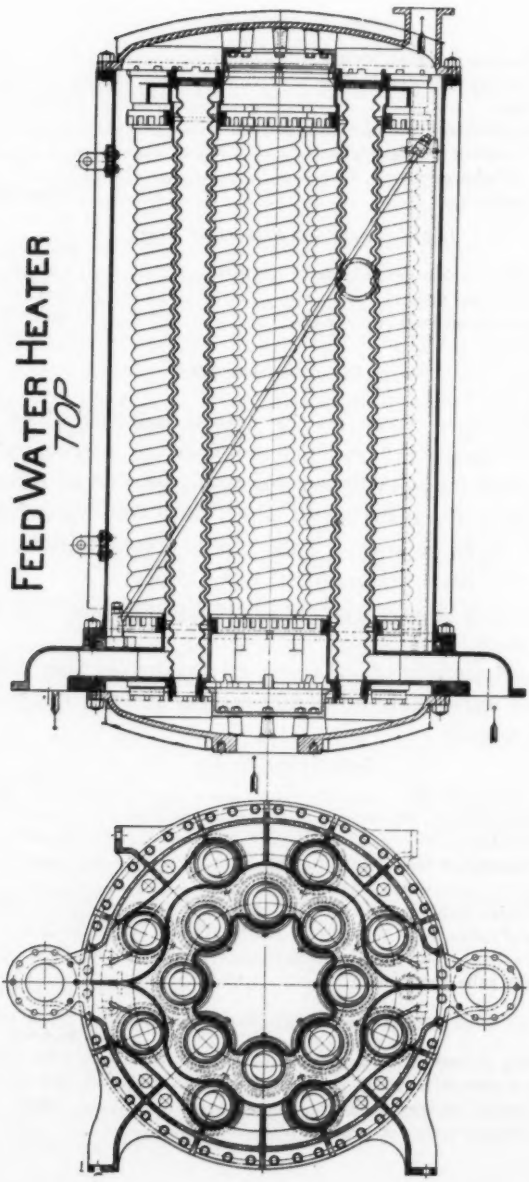


Plate VII.

AIR-COMPRESSOR PLANT.

On the center-line bulkheads, above the grating in the engine room, are nine 11 by 11 by 12-inch Westinghouse steam-driven air compressors, four being located in starboard engine room and five in the port engine room. The nine compressors have a combined capacity of 360 cubic feet of free air per minute at 150 pounds pressure per square inch. A working pressure of 100 pounds is maintained by an automatic governor on each compressor. The compressors in each engine room discharge in parallel into four air reservoirs, two in each engine room, with a capacity of about 45,000 cubic inches each. These reservoirs supply air to the following systems:

A line fore and aft along the center-line bulkhead below the grating in both engine rooms, branches from which lead to valves and hose connections in both engine rooms for pneumatic tools.

A line running to the workshop, with suitable hose connection.

A 2-inch line to firerooms with $1\frac{1}{4}$-inch take-off in each fireroom, for blowing soot off the boiler tubes, and $\frac{3}{4}$-inch outlets for pneumatic tools.

The gas-ejector system, for ejecting smoke from the turret and 5-inch broadside guns.

BOILERS.

The boilers on this ship are of the Babcock & Wilcox type, and were designed, built and installed by that boiler company. The battery consists of twelve boilers, placed four each in three watertight compartments, the boilers in all compartments being symetrically arranged.

Each boiler has one drum extending the full width of the boiler and is located within the casing so that about one-fourth of shell is in contact with the gases at the top of the combustion chamber. Immediately below the drum are the front headers, and at the back of the boiler are the back headers connected by tubes set at an angle of 15 degrees.

The weight of the drum and of the front headers, etc., is carried by the furnace front; that of the back headers by a light structural girder which also forms a portion of the casing. This is protected from the flames and heat by the bridge wall of brick extending the full width of the furnace. The side walls are rectangular water legs which extend from the lower row of tubes to ash pans in the front and at the back to the surface of the grate. Each of these side walls is connected to a round drum carried in the boiler front immediately below and connected to the front headers, providing in this way for the constant circulation of water, at the same time allowing means of blowing off and of drainage.

The boiler casings are made of steel plates and shapes, fire bricks and block magnesia being used for insulation. Front and back at the header spaces there are doors of such size that they are easily removable for access to the water sides of the boiler tubes. On the side, removable section and doors give access to the baffle plates and fire sides of the boiler tubes.

Each furnace is provided with five furnace doors, and also four ashpit doors, all opening inward to the furnaces and ash pans and overbalanced so as to be automatic in case of a sudden outrush of steam. The grate bars are of the ordinary common type, of such size and shape as to be easily handled by one man.

Between each of the five furnace doors is a rectangular cast-iron frame or air box as deep as the furnace front, making four in all for each boiler. Into each of these cast-iron frames are fitted two burners of the Schutte-Koerting flat spraying type. These burners vary in size from 1 m/m to 1½ m/m, and at 200 pounds pressure will atomize from 145 to 300 pounds of heated fuel oil per hour.

Boiler drums.—The boiler drums are of open-hearth steel plate, and have two seams which are butt-strapped inside and out. The heads are convex, having a radius equal to the inside diameter of the drums, and in each is a flanged manhole shaped from the same plate. All butt straps were found to be of correct curvature of the drum under hydraulic pressure, the rivet holes being punched smaller than the diameter of the rivets

to be used and then drilled out to full size after the plates were rolled and assembled. Wrought-steel pads, securely riveted to the drums, are provided for attaching all valves and fittings:

The following internal fittings are provided.

1 6-inch dry pipe;
1 2½-inch main feed pipe;
1 2½-inch auxiliary feed pipe;
1 1½-inch pipe and scum pan for surface blow;
Zinc protectors and buckets.

The following external fittings are provided for each drum:

1 5½-inch steam stop valve;
1 triple safety valve—each valve 4 inches in diameter;
2 2½-inch stop valve for main and for auxiliary feed;
2 2½-inch check valve for main and for auxiliary feed;
2 2-inch water gages of composition;
1 1½-inch surface-blow valve;
1 steam gage;
3 gage cocks;
1 air cock.

Other boiler fittings provided are:

1 drain cock;
1 salinometer pot connection;
1 connection for testing water in boilers;
2 steam tube cleaners with metallic hose.

Headers.—The headers are of forged steel, forged in one piece into a rectangular section, having a male and female sinuous side for nesting when assembled. The front side is provided with numerous but regularly spaced handholes to provide access to the water sides of all the boiler tubes. The covers for these handholes are of pressed steel, and held in place over the gasket by one stud, nut and dog. All holes for the generating tubes and the nipples connecting headers with the drums are bored and reamed to the size of the boiler tubes to be used.

Tubes.—The generating tubes are straight and made of seamless cold-drawn steel, as are the header and drum connecting nipples.

Data for Boilers.

Number of boilers	12
Length of drum, feet and inches	19-00¾
Inside diameter of drum, inches	42
Thickness of drum plate, inch	00$\frac{21}{32}$
Heating surface per header and tubes, square feet	174
Total heating surface for one boiler, square feet	5,353
grate surface for one boiler, square feet	119
H.S. for all boilers, square feet	64,234
G.S. for all boilers, square feet	1,428
Ratio H.S. to G.S.	44.93
Number of boilers per smokepipe	6
Height of smokepipe above grate, feet and inches	92-01¼
Area through smokepipe, square feet	101.56
Ratio G.S. to smokepipe area	07.03
Size of generating tubes, inches	2 and 4
Total number of headers per boiler	31
Thickness of 2-inch generating tubes, B.W.G	8
4-inch generating tubes, B.W.G	6
Exposed length of generating tubes, feet and inches	8-00
Number of 2-inch generating tubes per boiler	1,100
4-inch generating tubes per boiler	31
Inclination of tubes, degrees	15
Height of boiler, feet and inches	13-10¼
Length of boiler, feet and inches	9-01½
Width of boiler, feet and inches	20-02¼
Weight of one boiler, complete, dry, pounds	106,966
water in one boiler, center of drum, pounds	20,420
one boiler, complete, center of drum, pounds	127,386
Total weight of boilers, dry, pounds	1,283,592
water in boilers (center of drum, 58° F.) lbs	245,040
(full, 58° F.), pounds	312,996
boilers (full), pounds	1,596,588
Weight of boiler, dry, per cubic foot, pound	4.18
wet, per cu. ft., lbs. (middle cock, 58° F.)	4.99
(full, 58 degrees F.)	5.21
Ratio of weight of boilers dry to boilers wet (middle cock, 58 degrees F.)	1:1.19
Designed working pressure, pounds per square inch	200
Test pressure, pounds per square inch	300

OIL-FUEL SYSTEM.

Designed as an auxiliary for use in emergencies is a fuel-oil system installed for burning oil in combination with coal.

The fuel oil is carred in double bottoms each of which has a vent pipe to the main deck. These bottoms are under engine

rooms, from where oil is pumped out by a pump in each engine room to the two cylindrical settling tanks. These pumps have a Macomb strainer on both the suction and on the discharge. The settling tanks, each of 3,750 gallons capacity, are located in watertight compartment, outboard of the forward ends of each engine room, and have in them steam-heating coils with suitable drains.

Forward of the engine rooms the suctions of the two settling tanks join into one main leading to the forward fireroom. In each fireroom two suctions lead to the two fuel-oil service pumps. On the delivery end of these two fuel-oil service pumps in each fireroom are two Schutte-Koerting corrugated-film oil heaters and two fuel-oil high-pressure strainers, which discharge to the four oil-fuel mains leading to the four boilers in that compartment. On each of these boiler oil-fuel mains are the eight burners in four air boxes.

Plate VIII illustrates the oil-fuel heater.

There are steam smothering valves to all the fuel-oil bottoms: C-90 and C-91 are controlled from the after fireroom and the others from the starboard engine room.

Data for an Oil Heater.

Weight empty, pounds	470
Number of tubes	1
Outside diameter of inner tube, inches	$0\frac{5}{8}\frac{1}{2}$
outer tube, inches	$0\frac{5}{6}\frac{4}{4}$
Thickness of tubes, B.W.G., inner	10
outer	9
Length of tubes between tube sheets, inches	48
Thickness of tube heads, inch	$00\frac{5}{16}$
Heating surface tubes, square feet	$10\frac{1}{2}$
total, square feet	12

Data for Official Four-hour Trial.	Firerooms.		
	1	*2*	*3*
Temperature of entering oil, degrees F.	83	83	83
discharge oil, degrees F.	151	178	189

Capacity per hour of one heater, 6,000 pounds of oil from 70 degrees F. to 280 degrees F., with steam at 200 pounds' pressure.

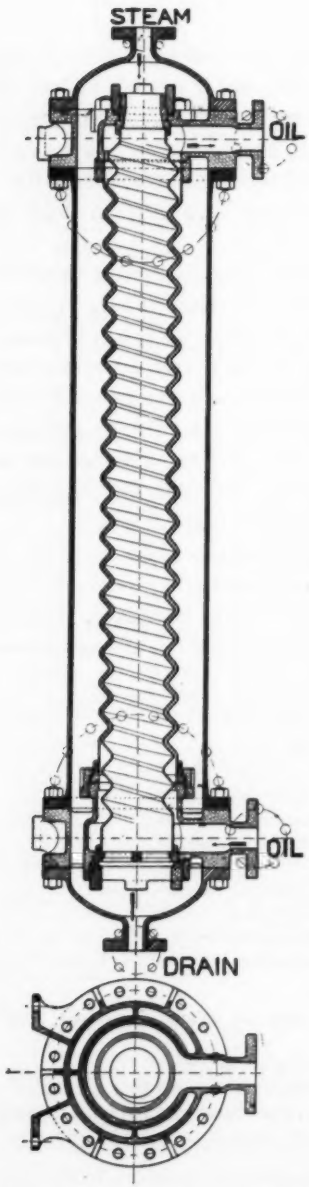


Plate VIII.—FUEL-OIL HEATER.

MAIN AND AUXILIARY STEAM LINES.

The two main steam pipes are arranged symmetrically, one on each side outboard of the firerooms. The branches from each boiler are 5½ inches in diameter and the main steam lines on each side of the ship, beginning in the forward firerooms, are 7½ inches in diameter, increasing in the middle fireroom to 9½ and in the after fireroom to 11 inches at the forward boiler, 12½ inches at the after boiler and 14 inches from there to the engine rooms. From the forward engine-room bulkhead steam passes to separators, from which point branches are taken to the turbines as shown in Plate I.

In the firerooms there is no auxiliary steam line, but the main steam lines port and starboard are cross-connected by a 3-inch pipe in each fireroom, from which connections are taken for auxiliaries therein.

In the forward fire-room there is a 7½-inch cross connection along the forward bulkhead with valves and castings, from which branches lead to the dynamo rooms, galley, whistle, forward heating system and forward auxiliaries.

From the separators in the engine rooms there is a 7½-inch pipe passing aft through each engine room and through the middle-line bulkhead, which forms a connecting loop aft between the two sides of the ship.

The main and auxiliary steam pipes are so arranged that they drain by gravity either to the separator or boilers. Where pockets necessarily occur the pipes are drained by traps.

All valves in the main steam line have by-pass valves attached to relieve them when jammed on their seats.

All steam pipes above 2 inches are of seamless-drawn steel tubing with rolled-steel flanges, the working pressure being 210 pounds per square inch.

The expansion of the main steam pipes in the engine rooms, the firerooms, dynamo rooms and pipe passages are taken care of by expansion joints in each compartment; where pipes are bolted to athwarpship bulkheads proper allowance has been made for expansion by making the bolt holes oval.

The piping is lagged with sectional magnesia covering with canvas sewed on and well painted and lagged with galvanized sheet iron.

AUXILIARY EXHAUST LINES.

The auxiliary exhaust lines, which are of copper, lead on each side from the forward fireroom aft on each side to each engine room through the pipe passages, joining in the forward fireroom with branches leading to the dynamo room, the anchor engine and the forward ice machine. In the after fireroom a branch leads from the port side to the safety-valve exhaust pipe. Aft these two exhaust pipes join at the after ends of the engine rooms. Connections are provided to direct the exhaust steam into either main condenser, either auxiliary condenser, either feed heater, either H.P. ahead turbine, either L.P. ahead turbine or into the atmosphere.

ASH EJECTORS.

In each fireroom there is an independent ash-ejector system of the Horace See type, the hoppers being outboard at the bulkhead on each side. From the hoppers the ash-discharge piping leads upwards overboard through the armor above the water line with but one bend; the latter and the method of securing the shell-bend pieces being of the Bureau pattern.

Each pair of hoppers is supplied by one double-acting vertical Warren steam-driven pump 12 inches by 7 inches by 12 inches. These pumps have but the one discharge, and they are located on the stanchions at center line of each fireroom.

AUXILIARY FEED PUMPS.

On the starboard side of the stanchions on the amidship line in each fireroom is located one Blake vertical-plunger, double-acting, single, center-packed, auxiliary feed pump; size of pump, 14½ inches by 9½ inches by 18 inches.

There are for these three pumps no salt-water suctions. A 2½-inch hose connection is fitted for pumping out boilers. All have grease extractors.

FIREROOM FIRE AND BILGE PUMPS.

On the port side of the stanchions on the amidship line in each fireroom is located one Blake, vertical-piston, double-acting, single, fireroom fire and bilge pump; size of pump, 12 inches by 10 inches by 18 inches. These pumps have no fresh-water suctions.

PUMPING OUT BOILERS.

The bottom-blow connections to boilers have a 2½-inch hose connection for use in pumping out boilers by means of the auxiliary feed pumps.

ASH HOISTS.

The ash-hoist engines were designed and built by the Hyde Windlass Co., and are located in the boiler-room uptakes, the ventilating trunks containing the bucket guides, cables, sleeves, etc., and are operated from the main deck. The bucket-guide rails extend from the bottom of the ventilation trunk to a point well above the main deck. There are no trolleys for taking ashes from ventilators to ship's side. On the tests with 100 pounds steam pressure 300 pounds were hoisted from the fireroom floor to the upper deck in seven seconds, including starting and stopping.

The hoisting engines are of the reciprocating type, with a rope drum, fitted with a follow-up and reversing gear, and an adjustable safety gear to prevent overriding and which stops the engine when the ash bucket reaches the fireroom floor.

Number of cylinders	2
Diameter of cylinders, inches	4½
Stroke, inches	4½
Diameter of drum, inches	10
Number of ash hoists in ship	3

FORCED-DRAFT BLOWERS.

In each fireroom on a level with the upper platform deck, are located four electrically-driven forced-draft blowers. General Electric motors are fitted with Sirocco fans 30 inches

in diameter. They are located in pairs on each side of the ship in airtight compartments, and are supplied with air from the uptake spaces immediately above, or from one ventilator, and deliver directly into the top of their firerooms.

Their speed on the full-power trial averaged 621 revolutions per minute at 125 volts, and 1,144 ampères, the total average horsepower being 220. The control is either at the motors or the fireroom floor plates.

Voice tubes lead to the central station and the firerooms directly below.

DESCRIPTION OF THE FIRE MAIN.

The fire main in the engine and firerooms is a steel, lead-lined pipe, running on each side of the ship, continuously, through each engine room, steam-pipe passage and fireroom, under the protective deck. Six-inch flooding valves and piping are fitted for all lower inboard and outboard bunkers, which may be operated from firerooms or from the ammunition passages, and from compartments B-100 and B-101, on the berth deck.

Forward from the firerooms, the fire main continues through wing passages on each side to frames 26, where there is a cross connection. A 4-inch branch leads forward on the starboard side to the forward pump room. One "fire and bilge" pump discharge pipe connects to each fire main in each fireroom, by means of a 5-inch pipe branching to both sides; these fireroom pumps also connect to the flushing system, by separate valves and pipes in compartment B-100 (ammunition passage), at frames 53, 63 and 71. Six-inch valves and pipes connect the fire main to both distiller pumps in the engine rooms at frame 80, and 6-inch valves and pipes connect the fire main to the engine room "fire and bilge" pumps in both engine rooms.

Aft the fire mains run outboard and aft through wiring and wing passages, and join amidships, between frames 110 and 111, with a hose-connection branch leading aft to steering-gear compartment, D-35, frame 115.

The following is a list of connections to fire plugs, etc.:

Compartment.	Frame.	Size.	Remarks.
		inches.	
A-100	14	4	Relief valve set at 100 pounds.
A-49	22	1¼	To torpedo air compressors.
A-51	26	6	Connection to sprinkler system.
A-52-S	25-26	4	Riser to plugs, Nos. 27-44-57-58 and forward relief valve.
A-52 S	25-26	4	Riser to plugs 15-29-45-59.
A-52 P	25-26	4	Riser to plugs 14-28-46-60.
A-52 S	25-26	2½	Hose connections.
A-52 P	25-26	2½	Hose connections.
A-52 S	25-26	6	Forward branches join—cut out.
A-52 S	25-26	6	Forward branches join—cut out.
A-52 P	25-26	4	Cut out, leading forward.
A-52 S	26	6	Cross connection—cut out.
A-52 P	26	6	Cross connection—cut out.
A-66 P. and S.	43	2½	Hose connection to 5-inch ammunition passage.
Dyn. Rm. Trunk	46	4	Cut out to risers.
B-100	49	2	Admiral's, captain's and chief of staff's bath rooms.
B-1 P. and S.	50	2½	Cut out to risers.
B-100	60-61	3	Firemen's wash rooms, galley, bakery and ash chutes.
B-2 P. and S.	61	4	Cut out to risers.
B-100	68	1½	Crew's galley.
B-3 P. and S.	71	3½	Cut out to risers.
C-1	85-90-93	6	Flooding.
C-2	80-81	5	To sprinkler system.
C-2	85-90-93	6	Flooding.
C-1	88	3½	Cut out.
C-2	88	3½	Cut out.
D-53 S. and P.	110-111	6	After branches joined—Cut out. Sprinkler system.
Wiring passage, D-59.	108	5	Riser to flushing system.
Wiring passage, D-52	103	2½	Cut out to riser.
Wiring passage, D-60.	108	2½	Cut out to riser.
D-54	111	3½	To hose connections in steering gear compartment D-35.
D-111	107-180	4	Relief valve, lift 100 pounds.

A 5-inch valve and pipe connects fire main with flushing system at frame 46, berth deck, port, compartment A-110. Starboard and port engine room "fire and bilge" pumps connect with flushing system at frame 80, compartment B-100. There is a branch to fuel-oil filling pipe on the gun deck at frames 80 and 81.

Salt water for admiral's, captain's, chief of staff's and executive officer's baths: valves located at frame 49, compartment B-100 (ammunition passage). Salt water for crew's galley

turns on at frame 68, compartment B-100. Salt water for firemen's wash rooms is taken from the flushing system, between frames 60 and 61, in compartment B-100; its valve being fitted on gun deck. Branches lead to officers' galley, bakery, and portable ash chutes.

Fire-main relief valves are fitted between frames Nos. 107 and 108 berth deck, D-111 starboard, and at frame No. 14, in compartment A-100. They are set at 100 pounds, and their discharges lead overboard via scuppers.

DESCRIPTION OF DECKS, BRIDGES, ETC.

Masts.—There are two masts of the cage or basket type, located at frames 49 and 67, on which are located searchlight platforms, portions of the wireless telegraphy outfit, signal yards, etc.

Bridges.—There are three bridges: steering platform, flying bridge and signal bridge, the latter being the lowest. There are no after bridges.

Forecastle Deck.—This deck extends from the stem to frame 52. On its after part is a deck house containing the admiral's and captain's quarters. On its level are skid deck beams, in which are stowed all boats except the life boats, which hang from davits aft and the two fifty-foot steamers, which stow on the main deck.

Main Deck.—The main deck extends from stem to stern. Forward of frame 52 is the wardroom country. Abaft this frame, situated amidships, are deck house containing the crew's galley, officers' galley, butcher shop, bakery, blacksmith shop and foundry and two boat cranes, one on each side at frame 63. The crane towers also serve as additional searchlight platforms.

Gun Deck.—The gun deck is also continuous and the full width of the ship. It has within it forward of the casemate the quarters for warrant and for junior officers, the latter being on the starboard side. Within the casemates are located the four firemen's wash rooms, coaling-gear machinery rooms, fire-room elevators and general mess store and issuing rooms.

Abaft and outside the casemates to frame 106 are the laundry and the offices of the following: Paymaster, executive officer, engineer, and ordnance, sick bay and medical offices, armory; and on the port side, the post office, chief petty-officers' quarters, servants' wash room, ship's store, ordnance storeroom, pantry, printing office and rooms of the chief master-at-arms and the sergeant of marines. Abaft frame 106 are the crew's spaces, wash rooms and water closets.

Berth Deck.—The arrangement of this deck, beginning forward, is as follows: Paint room, chain lockers, junior and warrant officers' storerooms, general store-issuing room, crew space frames 25 to 46. From frames 35 to 46 on the starboard side are located mine-defense outfit, ice-machine room, paymaster's issuing room and clothing and small stores; on the port side are prisons and detention cells. From frames 46 to 77, outboard on either side, are the upper coal bunkers: amidships are the uptake spaces, cold-storage compartment, ice-machine rooms, evaporator room, machine shop. Abaft frame 95 to frame 122 are crew spaces. From frame 122 to the stern is the lumber and pipe storeroom.

Protective Deck.—The space under the berth deck on the slope at the ship's sides is divided into numerous compartments and used for various stores, excepting the space between frames 19 and 112, which is a continuation of the upper platform deck.

Upper Platform Deck.—This deck extends from frames 19 to 112, and on it from forward aft are located stores, windlass engine, magazines, handling rooms, central station, blower rooms and coal bunkers.

Lower Platform Deck.—This deck extends from bow to stern. From frames 3 to 11 are the forward trimming tanks. From frames 11 to 25 are sails and awnings, storeroom and torpedo room. From frames 25 to 46 are magazines and dynamo rooms. From frames 46 to 71½ are the boiler compartments. From frames 71½ to 80 are magazines and storerooms. From frames 80 to 95 are the engine rooms and storerooms. From frames 46 to 71½ on either side, outboard, are the lower coal bunkers. From frames 95 aft to 112 are maga-

zines and storerooms and the steering-engine room. From 112 aft are the sanitary-pump room and the steering-gear rooms.

Hold.—Between frames 3 and 11 are the forward trimming tanks; 112 aft are the after trimming tanks; frames 11 to 19, fresh-water tanks; 19 to 46, dynamo room and storerooms; 46 to 72½, boiler compartments and coal bunkers; 71½ to 80, storerooms; 80 to 95, engine rooms and storerooms; 95 to 112, shaft alleys and storerooms.

Double Bottoms.—The double bottoms extend from frames 11 to 112. In ten of these, under the boiler compartments, is carried reserve feed water; under the engine room are ten fuel-oil double bottoms. All other double bottoms are dry.

VENTILATING SYSTEM.

Artificial ventilation has been provided where necessary for all quarters, living spaces, passages, storerooms and magazines. There are thirty-two ventilating motors and fans, each on its own circuit, for supplying air on the plenum system to the different compartments, the air being renewed there at a previously-determined rate according to the necessities entailed by its use. These electrically-driven fans are manufactured by the American Blower Co., Detroit, Mich., and are located as shown in the table on following page.

The air as supplied by these fans, running at their maximum speed, has a pressure of from 4 to 9 pounds to the square foot and a velocity of from 1,200 to 7,000 feet per minute, depending on the designed capacity and the purpose for which the compartment is used, and these blowers on tests averaged about 75 per cent. of their rated capacity.

None of the principal longitudinal or transverse watertight bulkheads of the ship have been pierced by ventilation ducts. No ducts have been carried through the transverse slopes of the protective deck, but in all cases through the flat and the fore-and-aft slopes. Where ducts pass through protective deck they are made watertight to the highest practical point above the berth deck. All ducts passing through magazines are galvan-

No.	Compartment.		Deck.	Side.	Frame.	Remarks.
	No.	Name.				
1	A-118	Lamp room.....	Main.	P.	4-5	
2	...	Turret No. 1.....	...	...	...	On upper handling-room level.
3	...	Turret No. 2.....	...	...	...	do
4	B-116	.....	Gun.	P.	44	...
5	B-116	Crew space.....	Gun.	S.	44	...
6	B-116	Crew space.....	Gun.	S.	47	...
7	B-116	Crew space.....	Gun.	P.	58-60	...
8	B-116	Crew space.....	Gun.	P.	69-70	...
9	C-113	Crew space.....	Gun.	S.	74	...
10	C-113	Crew space.....	Gun.	P.	78	...
11	C-113	Crew space.....	Gun.	S.	78	...
12	...	Turret No. 3.....	...	...	...	do
13	C-113	Crew space.....	Gun.	S.	79	Magazine cooler.
14	C-113	Skylight trunk.....	Gun.	A.	91	...
15	...	Turret No. 4.....	...	...	...	Upper H. R. level.
16	D-109	Crew space.....	Berth.	P.	104	...
17	...	Turret No. 5.....	...	...	...	do
18	A-104	General stores.....	Berth.	P.	20-21	...
19	A-108	Crew space.....	Berth.	S.	35	Magazine cooler.
20	A-108	Crew space.....	Berth.	P.	32	...
21	A-110	Crew space.....	Berth.	S.	35-36	...
22	C-100	Crew space.....	Berth.	P.	83	...
23	C-100	Crew space.....	Berth.	S.	83	...
24	D-109	Crew space.....	Berth.	P.	104	...
25	D-109	Crew space.....	Berth.	S.	102	Magazine cooler.
26	D-109	Crew space.....	Berth.	S.	101	...
27	D-113	Crew space.....	Gun.	S.	114	...
28	D-112	Lumber and stores...	Berth.	S.	114	...
29	C-43	Blower room.....	U.P.	S.	83	...
30	C-44	Blower room.....	U.P.	P.	83	...
31	C-43	Blower room.....	U.P.	S.	84	...
32	C-44	Blower room.....	U.P.	P.	84	...

ized-steel seamless tubing, or built of heavy-gage steel worked watertight. A natural-exhaust duct, equal to the area of the mechanical-supply duct, has been provided for each magazine, and located as far as practicable from the supply duct. The upper ends of these ducts are carried up close to the gun deck and terminate there with a gooseneck, the lower end of which is bell mouthed and covered with wire mesh.

McCrary adjustable elbows, fitted with butterfly dampers, have been used for supply terminals in all quarters, living spaces and elsewhere. In the quarters they are nickel plated and elsewhere galvanized. All openings of these elbows are fitted with portable wire mesh.

There are adjustable mushroom cowls which fit in the deck and serve as inlets for the various blowers. Two forward and one aft are the only ones fitted to unship at "general quarters."

The ventilation of the bunkers is exhaust only, and that not forced. Galvanized-steel tubing leads from all bunkers into the uptake spaces and are fitted with inswinging airtight doors of the damper type; all openings into the bunkers are fitted with wire mesh.

For ventilating the engine rooms there are two blowers, port and starboard. Branches are led from the mains to all working stations, platforms, air casings and corners of the rooms where required for efficient ventilation.

Sheathing.—To retard transmission of heat to adjacent magazines and ammunition passages the pipe passages, firerooms, steering-engine room, engine rooms and coal-bunker bulkheads separating the former from the five latter are lagged or sheathed.

From the forward lower coal-bunker bulkheads exhaust piping leads to the air casing under the protective deck in the forward fireroom, whose air in turn exhausts to sheathing in the fireroom hatch, and from there to the atmosphere. The heated air from the casing under the protective deck of the other two firerooms exhausts similarly, as does the sheathed after athwartship bulkhead of the after fireroom.

The under sides of the protective deck, within the fire and the engine rooms, have been fitted with sheathing, with an air space. This space within the engine rooms is connected with the ventilating system for supply, and to the engine hatches for exhaust. In the firerooms this space is not fitted for supply except forward and aft the forced-draft blowers; the exhaust goes into the uptakes near the top.

Heated Air.—A new departure in connection with the ventilation of the living quarters is the installation of steam-heating coils in the air-supply pipes. These heating coils in cold weather will raise the temperature of the incoming new air, thereby relieving the heating coils in the living quarters of a good deal of the work formerly required of them.

HEATING SYSTEM.

The heating system is divided into two sections, one forward and one aft. The forward section is divided into six circuits, one of which is for galley, bakery, and firemen's washrooms; two are for bath and pantry circuits, one for radiator circuit in admiral's and captain's quarters, and two for heater or thermo tanks located on berth and gun decks.

The after section is divided into six circuits consisting of one radiator, one washroom and sick-bay bath, and four thermo-tank circuits.

All steam lines to heating systems are fitted with pressure gages, stop and relief valves, set at 30 pounds; from stop valves the steam is carried to distributing manifolds and divided into circuits, each circuit being arranged with a stop valve so that it can be operated independently of the others.

There being no auxiliary steam line in firerooms, all the forward circuits receive their steam from the main steam line direct. The gravity-tank heater coils receive steam from the forward circuit. Steam for the laundry is taken from the after circuits.

The drains from radiators and thermo tanks are also divided into circuits and lead to receiving manifolds, located forward and aft, each circuit being arranged with check and stop valves to prevent water from one circuit backing up into another. The discharge from these manifolds leads to a steam trap in the port engine room, which discharges into the filter tank, or to main and auxiliary condensers.

The drains from galleys and pantries also lead to traps in port engine room, discharging into filter tanks and the condensers. The drains from bath rooms and other water heaters have no connection with radiator drains but go direct to auxiliary exhaust lines in Nos. 1 and 2 firerooms and in both engine rooms.

The heater-circuit steam and drain pipes consist of seamless-drawn brass pipe, iron-pipe size; all pipes up to 1½ inches being connected together by composition fittings, but above that size

composition flanges are used. Connections at watertight bulkheads are made with composition stuffing boxes, and U bends are installed throughout to provide for expansion.

The radiators consist of coils made up of 1-inch seamless-drawn brass pipe, iron-pipe size, composition fittings, stop and air valves.

The number of heater or thermo tanks fitted throughout the berth and gun decks are 21, and consist of casting or header with 1-inch seamless-drawn brass-pipe coils, varying in number from 7 to 22, according to size of box. These boxes are of the vertical and horizontal types, and fit between deck beams, or in the case of the larger size stand on a frame on the gun deck. These boxes are fitted to receive air from ventilating fans and pass it through the steam-filled coils before it discharges into living spaces. The coils are insulated with black magnesia, covered with No. 16 sheet iron. By arrangement of dampers the air may be by-passed in warm weather. In case of a leak any coil in box can be removed without disturbing the others.

DRAINAGE SYSTEM.

The following is a genral description of the drainage system, the features covered being as follows:

1. 15½-inch main drain.
2. 5½-inch secondary drain.
3. 4⅝-inch double-bottom drain.
4. 4½-inch forward bilge drain.
5. 4⅝-inch after bilge drain.
6. 5-inch independent drain fireroom (Bu. of S.E.).
7. 5-inch independent drain engine rooms (Bu. of S. E.).

Main Drain.—It is 15½ inches in diameter throughout its entire length and is located on the starboard side outboard. It extends from the after end of the forward fireroom to the forward end of the engine room, where it branches and runs athwarthships, each branch connecting to a centrifugal pump.

A 15½-inch stop-check valve is located in each fire and engine room, forming part of the main system, and all are

operated either at the valves or on the berth deck through deck plates.

In the port engine room, at frame 84, there is a 5-inch connection between the main drain and a C. & R. manifold, located at frames 83–84. This manifold is connected to the fire and bilge pumps in the fire and engine rooms.

Secondary Drain.—It is 5½ inches in diameter throughout its entire length, frames 50–92, and has connections to all fire and bilge pumps through C. & R. and S. E. manifolds; also to handy-billy-pump manifolds, located on upper platform deck, forward at frame 46 and aft at frame 95. The forward connection is made directly to the secondary drain and the after connection is made through a C. & R. manifold located in the starboard engine room at frame 91.

The forward end of the secondary drain connects to the 4½-inch forward-bilge drain, located in forward fireroom, frame 50–51, and at the after end to C. & R. manifold in starboard engine room, frame 91.

There is a 5½-inch connection from the bilge wells in each fire and engine room to the secondary drain.

Double-Bottom Drain.—It is on the port side of the firerooms, between frames 46 and 71, and is divided into three sections, each being 4⅝ inches in diameter where suctions to independent double bottoms are 3 inches in diameter. It drains twelve double bottoms and has a connection to one C. & R. manifold in each fireroom.

The double-bottom compartments are flooded through S.E. manifolds connected to C. & R. manifolds by operating their respective stop-check lift valves.

Oil Double Bottoms.—The double bottoms for oil-fuel stowage are under the engine rooms and have independent suctions to the two oil-fuel pumps, one in each engine room.

Forward-Bilge Drain.—It is 4½ inches in diameter and drains all the double bottoms forward of frame 41 and compartment A–11. The latter is the drainage tank for chain lockers, handling rooms, forward torpedo room and Nos. 1 and 2 turrets. This drain ends at frame 19 in a manifold,

from which suctions are led to the forward compartments and the triming tanks.

After-Bilge Drain.—The $4\frac{5}{8}$-inch after-bilge drain connects with the secondary drain, C. & R., at the manifold, frame 91, located in starboard engine room, draining the "D" compartments and the after trimming tanks, where it is 3 inches in diameter.

Independent Drains.—These drains come under the cognizance of Bureau of Steam Engineering. In each fireroom they are $4\frac{1}{8}$ inches in diameter and run from the bilge wells to the fire and bilge pumps.

The independent drains in each engine room are 5 inches in diameter. They drain the bilge wells in shaft alleys and engine rooms. The drains in the shaft alleys connect to C. & R. manifolds, in port and starboard engine rooms, frames 80, and from there have a direct connection to Bureau of Steam Engineering manifold, which is connected to fire and bilge pumps in the engine rooms.

C. & R. Manifolds.—Drainage System.

Location, frame.	Port or starboard.	Compartment.
19	Port	A-8
52	C.L.	B-1
57	C.L.	B-2
68	Port.	B-3
84	Starboard.	C-1
84	Port.	C-2
96	Starboard.	D-5
96	Port.	D-4
46	Starboard.	A–64 S.

TRIMMING TANKS.

There are four trimming tanks, two forward and two aft.

The forward tanks are flooded by a 6-inch sea connection at frame 11, starboard side. A 4-inch pipe is led to each trimming tank, where a 4-inch globe valve is operated at the valve and on the berth deck in deck plates, frame 11.

The after tanks are flooded by a 6-inch sea connection, frame 112, starboard side; a 4-inch pipe is led to each trimming tank, where a 4-inch globe valve is operated at the valve and on the berth deck, in deck plates, frame 112.

Capacities of Double-Bottom Compartments.

Compt.	Frames.	Side.	Cubic feet.	Tons F.W.	Tons S.W.
A-92	11-15	Across	1,594.8	44.5	45.83
A-93	15-19	Across	1,856.1	51.8	53.35
A-94	19-25	Across	3,373	94.1	96.91
A-95	25-31	Between wings	2,949.4	82.3	84.73
A-96	31-35	Between wings	2,136.1	59.6	61.38
A-97	35-41	Between wings	3,383.5	94.4	97.21
A-98	41-46	Center to port wing	1,881.4	52.5	54.07
A-99	41-46	Center to stbd. wing	1,881.4	52.5	54.07
B-80	46-50	Port	687.8	19.2	19.78
B-81	46-50	Starboard	687.8	19.2	19.78
B-82	50-55	Port	924.6	25.8	26.57
B-83	50-55	Starboard	924.6	25.8	26.57
B-84	55-59	Port	716.8	20	20.6
B-85	55-59	Starboard	716.8	20	20.6
B-86	59-63	Port	716.8	20	20-6
B-87	59-63	Starboard	716.8	20	20.6
B-68	46-50	Port wing	1,182.5	26.2	27
B-69	46-50	Starboard wing	1,182.5	26.2	27
B-70	50-55	Port wing	1,571	35.9	37
B-71	50-55	Starboard wing	1,571	35.9	37
B-72	55-59	Port wing	1,351.6	31.1	32
B-73	55-59	Starboard wing	1,351.6	31.1	32
(Wing double bottoms B-68 to B-73, inclusive, can be filled only to upper manhole—completely filled they hold 51 tons more of salt water.)					
D-97	95-100	Port and starboard.	2,326	64.9	66.82
D-98	100-105	Port and starboard.	2,161	60.3	62.11
D-99	105-110	Port and starboard.	3,207.5	89.5	92.18

Capacities of Trimming Tanks.

A-1	Stern-6	Side forepeak	3,512.5	98.06	101
A-2	6-11	Side forepeak	5,533	154.4	159
D-8	112-118	Forward after TT	3,369.7	94.18	97
D-9	118-stern	After TT	1,111.1	31.07	32

(Note.—Fresh water, 34.8 cu. ft. to ton; salt water, 33.79 cu. ft. to ton.

Capacities of Reserve Feed-Water Tanks.

B-88	46-50	Port	874.5	24.4	25.13
B-89	46-50	Starboard	874.5	24.4	25.13
B-90	50-55	Port	1,093.1	30.5	31.42
B-91	50-55	Starboard	1,093.1	30.5	31.42
B-92	55-59	Starboard	874.5	24.4	25.13
B-93	55-59	Port	874.5	24.4	25.13
B-94	59-63	Starboard	874.5	24.4	25.13
B-95	59-63	Port	874.5	24.4	25.13
B-96	63-67	Starboard	1,663	46.4	47.79
B 97	63-67	Port	1,663	46.4	47.79

Capacity of Fresh-Water Tanks.

Tank No.	Compartment.	Between frames.	Location.	Side.	Capacity.	
					Gallons.	Tons.
A-3	A	11-15	Hold	Starboard ...	6,103	22.7
A-4	A	11-15	Hold	Port	6,103	22.7
A-5	A	15-19	Hold	Starboard ...	6,103	22.7
A-6	A	15-19	Hold	Port	6,103	22.7
Gravity Tanks.						
No. 1	...	...	Main deck.	Starboard.	1,000	3.7
No. 2	...	...	Main deck.	Port.....	1,000	3.7
Total,	.....	.....	.....	.....	26,412	98.2

DECK DRAINS.

For sanitary purposes all drains from baths, toilets, wash rooms, pantries and galleys drain overboard directly through pipes by the shortest route, and are located as follows:

Name and use.	Frame.	Side.	Drain.	Scupper.
			inches.	*inches.*
Ward room and warrant officers' baths and toilets	11-12	Port	2½	5
Junior officers' baths and toilets.....	11-12	Starboard..	2½	4
Ward room, junior officers' and warrant officers' pantries.	22-23	Port	1½	3
Admiral's and captain's baths and pantries	48-49	Port	2	4
Firemen's wash rooms.....	56-57	Std. & port.	3½	8½
Officers' galley and bakery.....	56-57	Std. & port.	2	8½
Crew's galley.....	64-65	Std. & port.	2	8½
Petty officers' wash room.....	64-65	Std. & port.	2½	8½
Chief petty officers' and servants' wash room.....	94	Port	1½	2½
General mess pantry.....	90	Starboard..	1½	2½
Laundry	77-78	Starboard..	2	3
Sick bay	103	Starboard..	2	4
Crew's wash rooms and toilets.....	120-121	Port & std.	5	5½
Crew's wash rooms and toilets.....	123	Port & std.	5½	6

SANITARY AND FLUSHING SYSTEM.

The flushing system forward consists of a 5-inch steel, lead-lined pipe between frames No. 46 and 81, in compartments B-100 and C-100, forward of frame No. 46, reduced to 3 inches; a branch leads to forward ice machine, wardroom,

junior officers' and warrant officers' bath room. Branches lead to cabin bath rooms, to chief petty officers' and firemen's wash rooms, galleys, bakery, ash chutes. dish-washing room, sick-bay baths, laundry and fuel-oil filling line. This system is supplied by four engine-room fire and bilge pumps and the three fireroom fire and bilge pumps. A 5-inch bypass valve from fire main connects with this system at frame No. 46.

The flushing system aft consists of two electrically-driven centrifugal pumps located in compartment D-34, with a 5-inch discharge pipe leading up to crew's toilet and wash rooms on gun deck between frames No. 117 and 127. A 5-inch bypass valve from fire main connects with this system at frame No. 108, compartment D-113.

MAGAZINE FLOODING AND SPRINKLING.

There are three flooding systems for the three groups of magazines, forward, middle and after. Each system floods the magazines of its group on the lower platform deck, while the upper platform magazines are flooded or sprinkled by water from the fire main.

The forward and after systems are each composed of two separate systems, port and starboard, each of which is supplied by 8-inch sea connections, frame 31, forward, and frame 105, aft.

The middle system has its 9-inch sea connection between frames 72–73 and floods the middle group of magazines.

These flooding systems run above the lower platform beams forward and below, then, for the middle and after groups and branch pipes, are led up through the decks to valves in magazine spaces.

Adjacent to the sea connections are globe valves of corresponding size on the lower platform, operated at the valves on the berth deck in deck plates.

Upper-Platform Systems.—These are connected to the fire

main as described thereunder, and in other particulars have the characteristics of the normal flooding system. All magazine-flood valves are gate valves operated on the berth deck in deck plates.

Sprinkling Systems.—A sprinkling system is fitted in all magazines and shell rooms and consists in the magazines of 2½-inch brass pipe perforated on the under side, so that each powder tank can be sprinkled, and is fitted with cut-out valves in each magazine. The latter are operated at valve and on the berth deck in deck plates. This system is connected with the fire system which serves the flooding system on the upper platform.

MAGAZINE COOLING.

For all the magazines except those containing saluting powder and small-arms ammunition, provision is made for supplying artificially-cooled air. There are three systems supplying, respectively, the forward, the middle and the after group of magazines, each system being supplied by its own ventilating blower; the two forward blowers having a capacity of 2,500 cubic feet per minute and the middle 1,600 feet.

The method of cooling the air is as follows: Compressed air is led from one of the Allen dense-air ice machines to a Schutte-Koerting corrugated-film air cooler, where, on expanding, heat is extracted from the incoming air on its way to the magazines from the ventilating blowers.

Plate IX illustrates this air cooler.

Forward and aft the coolers are of three units and amidships of two units.

The magazines are lagged where necessary to prevent the conduction of heat.

The cooling capacity per unit is 800 cubic feet of air per minute from 70 degrees to 50 degrees F., when the dense air enters the cooler at 50 degrees F. and leaves at 30 degrees F. with sea water at 70 degrees F.

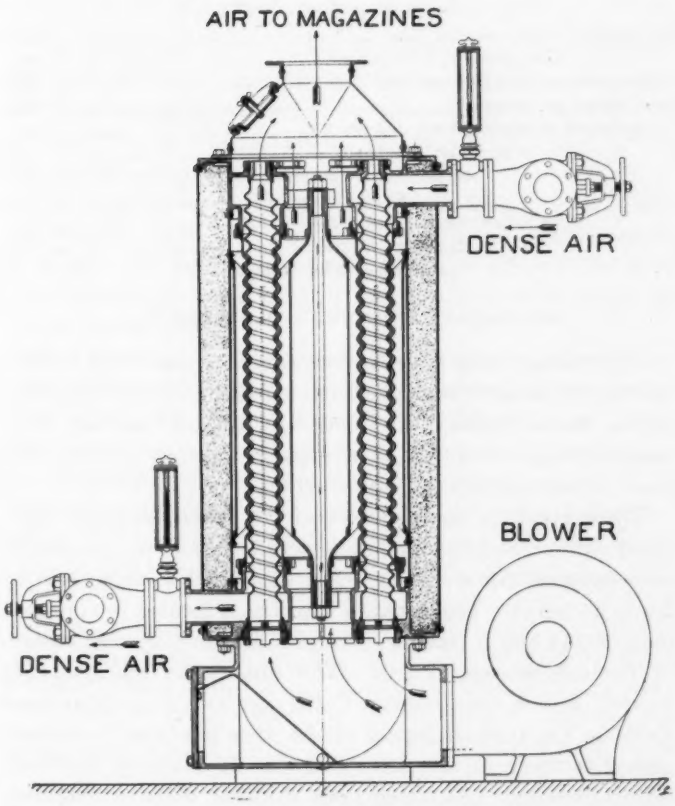


Plate IX.

Data of one Cooler.

Weight, empty, of a 2-unit cooler, in pounds	2,310
3-unit cooler, in pounds	3,350
Per unit number of tubes	6
cooling surface tubes, square feet	33$\frac{1}{2}$
headers, square feet	4
Diameter outside of inner tube, inches	03$\frac{7}{8}$
outer tube, inches	03$\frac{1}{8}$
Length of tubes between tube sheets, inches	37
Thickness of tubes, B.W.G	15
tube heads, inch	00$\frac{5}{16}$

Data for Official Trials.

Air pressure, high, pounds	254.5
low, pounds	72.8
Steam pressure at engine, pounds	85.7
Revolutions per minute	83.5
Temperature of expanded air, degrees F.	34.5
cooler inlet, degrees F.	25.2
outlet, No. 1, degrees F.	20.6
No. 2, degrees F.	19.2
at blower suction, degrees F.	53.1
discharge, degrees F.	72.1

(Average test for 13 days.)

REFRIGERATING PLANT (COLD STORAGE).

The refrigerating plant is located in compartment C-107, starboard side berth deck over engine room. The refrigerating rooms, five in number, are isolated by air locks and are insulated with cork covered on the inside with zinc, except the doors, which are covered with sheet lead.

There are four ice machines of the Allen dense-air type, fitted with re-coolers and water-jacketed compressor cylinder, each designed for a cooling effect equivalent to three tons of ice in 24 hours. The forward machine is located in compartment A-111 and is piped so it can be used on magazine coolers or the cold-storage system. The three after machines are located, one in compartment C-107 and two in compartment C-110. The cold-air piping of the four machines is so connected to the general system that any ice machine can work on cold storage, ice making tank, drinking butts or magazine coolers and return to re-coolers. Connections are fitted to air-compressor lines to build up pressures quickly in ice machines.

During the test of the plant the air was sent through cool storage, ice-making tanks and scuttle butts.

Machine Data.

Diameter of primer plunger, inches	02 11/16
steam cylinder, inches	10
compressor cylinders, inches	09
expander cylinder, inches	07 1/2
piston rods, inches	02
Length of connecting rod, center to center, inches	35
Stroke, inches	14

EVAPORATING AND DISTILLING PLANT.

The evaporator room is located amidships on the berth deck just forward of the engine-room hatches. In it are installed all the distilling plant except the four distillers and the two evaporator feed-water heaters, the six latter being in the port engine-room hatch.

The plant consists of four evaporators and four distillers, the designed capacity being 25,000 gallons of potable water in twenty-four hours, when running single effect. For purposes of economy the plant is also piped so that it can be run in double effect.

Evaporators.—The evaporator shells and heads are of steel plate, the steam-coil heads being of composition, with brass evaporating tubes arranged in pairs with return heads at the back end; these tubes are rolled into Muntz metal tube sheets at the front end, which is carried by the steam head.

Each evaporator is provided with steam gages and relief valves for the shells and the coils; there are also water columns, gage cocks, blow-off valves and salinometer pots.

Data for one Evaporator.

Diameter, feet and inches	5–06
Length between heads, feet and inches	6–02
Number of tubes	108
Size of tubes, inches	01¼
Length of tubes, feet and inches	5–11¼
Heating surface, square feet	312

Distillers.—The distillers are vertical and of the Schutte-Koerting corrugated-film type. Plate X shows this distiller in detail.

Data of one Distiller.

Weight, empty, pounds	864
Number of corrugated (film) tubes, copper lined	6
Shell is copper with composition flanges.	
Tube headers are composition.	
Water chests are composition.	
Thickness of tubes, B.W.G., inner	15
outer	15
Outside diameter of inner tube, inches	3$\frac{1}{32}$
outer tube, inches	3$\frac{7}{64}$
Cooling-surface headers, square feet	4
tubes only, square feet	36

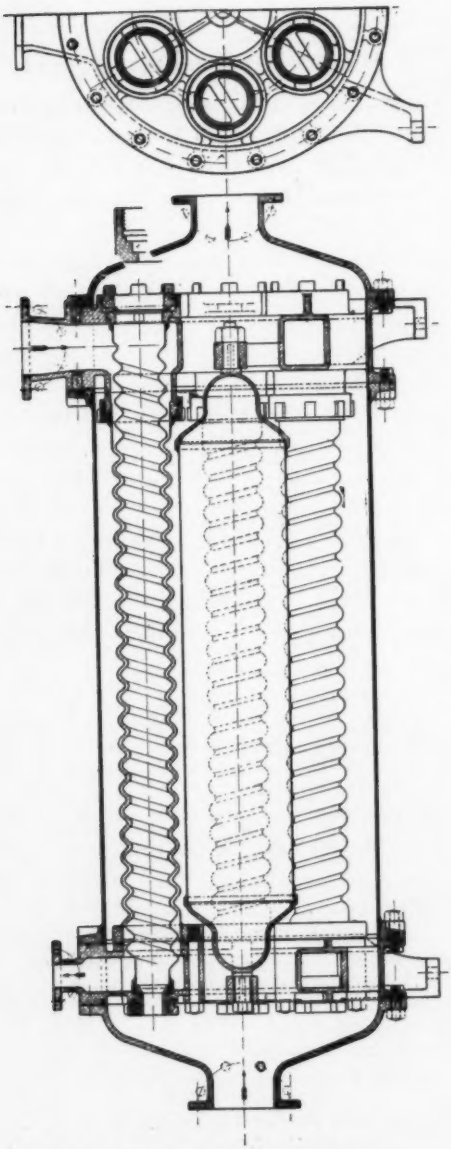


Plate X.—DISTILLER.

Data of Trial.

Gallons distilled per square feet of cooling surface per 24 hours...	161
in 24 hours (at rate of).....	27,030
Temperature of sea water, degrees F.....	71.3
evaporator feed water, degrees F.....	128.8
fresh-water discharge, degrees F.....	100.5
external air, degrees F.....	82
evaporator room, degrees F.....	124.5
highest, end of 4th hour, degs. F.	142
lowest, beginning of trial, degrees F.....	109
Concentration of water in evaporators, in 32ds.....	3.375
Quality of water distilled.....	Good.
Steam pressure in coils, average, pounds.....	57
shell, average, pounds.....	25.8
Pump strokes, evaporator feed pump.....	25.5
distiller circulating pump (both running).....	37.25

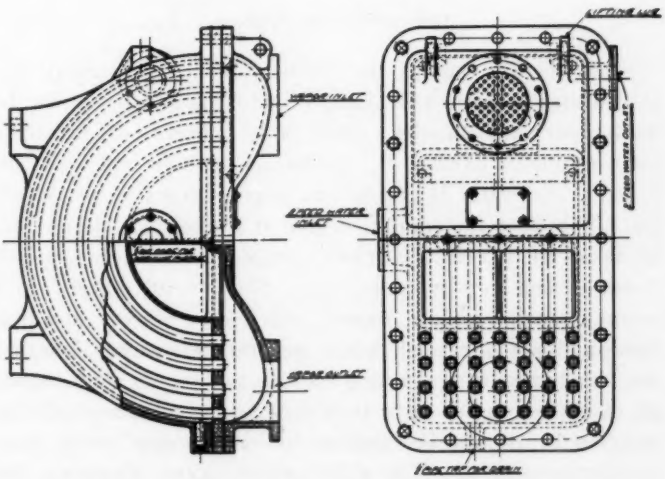


Plate XI.—EVAPORATOR FEED HEATER, U. S. S. "UTAH."

Pumps.—In the evaporator room are five Blake pumps, as follows:

2 Evaporator feed pumps;
2 Distiller circulating pumps;
1 Fresh-water pump.

Evaporator Feed-Water Heater.—The evaporator feed-water heaters are two in number and are of the Bureau curved-tube type and shown in detail on Plate XI. They take steam only from the dry pipes of the evaporator shell as it passes to the distillers. Sufficient distiller circulating water is forced through the heater and fed into the evaporator shell by the evaporator feed pump.

Separator.—For each pair of evaporators there is on the vapor line in the evaporator room a steam separator of the ordinary type. After the distilled vapor passes through the evaporator feed heater it passes through another separator of the ordinary type before it goes to its pair of distillers.

The designed capacity of one distiller is 262 gallons per hour.

FRESH-WATER SYSTEM.

From distiller pumps the fresh water is discharged into fresh-water tanks A-3, 4, 5 and 6 in forward hold, or may be discharged into laundry, chief petty officers' or firemen's wash rooms or overboard. From fresh-water tanks A-3, 4, 5 and 6 the water is pumped to gravity tanks located abaft and higher than the admiral's cabin at frames Nos. 44 and 45, by means of two electric-driven pumps placed in compartment A-46. From gravity tanks water flows to all officers' bath rooms, pantries, galleys, bakery, chief petty officers' and firemen's and servants' wash rooms, general mess pantry, laundry, sick bay, baths, battle dressing stations, and crew's wash rooms. In case the gravity tank is out of order all parts of ship supplied by it may be supplied by one electric pump from compartment A-46. The chief petty officers', firemen's and servants' wash rooms and laundry are fitted with storage tanks from which they take their supply. Other places are supplied directly from the fresh-water main or branches. For filling ship's tanks or reserve-feed tanks from water boats, etc., with fresh water, filling pipes are installed at frames Nos. 25 and 26, and 58 and 59. Each is fitted with by-pass to fresh-water main.

MACHINE SHOP.

The machine shop is located amidship on the berth deck between the two engine-room hatches and aft of the evaporator room. It is reached from the gun deck and through the port engine-room hatch. The following is a list of the machines installed, and each is driven by an independent, enclosed electric motor:

1 28-inch by 46-inch by 10-foot extension-gap lathe;

1 12-inch lathe, taking 18 to 24 inches between centers, mounted on bed of the above and to be readily removable therefrom;

1 14-inch screw-cutting lathe, 4 feet between centers;

1 15-inch column tool-room shaper;

1 28-inch upright drill;

1 16-inch sensitive drill;

1 universal milling machine;

1 12-inch double emery grinder;

1 portable cylinder-boring machine;

6 machinist vises.

The tools are provided with the most modern attachments, including scroll and drill chuck, index head, automatic cross feed, swivel table, pipe vises, etc., and all necessary tools, drills and cutters.

BLACKSMITH SHOP—FOUNDRY.

In addition to the machine shop there is furnished a steel blacksmith's forge with necessary tools and fixtures and with a 140-pound anvil. In the blacksmith shop, on the main deck abaft No. 3 turret, there is room for the ship's foundry, and possibly the coppersmith shop also.

ANCHOR WINDLASS.

The anchor engine is of the horizontal type and manufactured by the Hyde Windlass Co. It has two vertical shafts driven by a worm gearing direct from a worm on the crank shaft of the engine. Each shaft has on its upper end, above the forecastle deck, a wildcat, and the arrangement is such

that the wildcats can be operated together or independently of each other.

The windlass engine is a double engine, the diameter of the cylinders being 17 inches and the stroke is 14 inches. The working steam pressure is 150 pounds, but the engine is designed to stand full boiler pressure.

COALING ENGINES.

There are two steam-driven coaling engines manufactured by the Hyde Windlass Co., located at Frame No. 62 on gun deck on the port and starboard sides of the ship. These engines are of the vertical type, double cylinder, the diameter of the cylinders being 12-inch and the stroke is 10 inches. The working pressure is 150 pounds, but these engines are designed to stand full boiler pressure, and are fitted with governor gear to regulate the speed.

From each engine through miter gears a 5-inch shaft runs forward (its size there being 3½ inches), and to them are geared five winch heads, which can be thrown in or out at will by friction clutches. These shafts are cross connected by a shaft and miter gearing so that either coaling engine can actuate all winch heads.

OTHER POWER MACHINERY.

Of the latest types and conforming to the best commercial and Navy practice are the following machines:

1 motor-driven clothes-washing machine;
1 motor-driven mangler;
1 motor-driven wringer;
1 motor-driven ironer;
1 motor-driven potato peeler;
1 motor-driven dough mixer;
1 motor-driven meat chopper and grinder;
1 motor-driven ice-cream freezer;
1 motor-driven dish-washing machine;
2 motor-driven deck winches, 1 aft and 1 forward;
2 motor-driven fresh-water pumps forward;

2 motor-driven flushing pumps (salt water) aft;
2 motor-driven electric cranes, one on each side of the ship at frame 62, and fitted with suitable motor operating gear;
5 motor-driven turret F. D. blowers;
2 motor-driven torpedo air compressors;
2 motor-driven 5-inch ammunition conveyors;
20 turret-training motors;
10 turret upper-hoist motors;
10 turret lower-hoist motors;
10 turret elevating motors;
10 turret rammers.

ELECTRIC PLANT.

The dynamo plant is located forward of the firerooms, directly below the switchboard room and central station, all being under the protective deck.

The generator plant consists of four 300-k.w. turbo-generators, built by the General Electric Company. They are of the 6-pole compound-wound type, each capable of delivering at full load 2,400 ampères of current at 125 volts, with a speed of 1,500 r.p.m. They are driven by 2-stage Curtis turbines with three rows of moving buckets in each stage. The valve mechanism for controlling the speed is of the crosshead mechanical type, consisting of six valve units operated by an oscillating lever driven from the main turbine shaft through a double reduction-worm gear, and controlled by an inertia governor mounted on the primary-gear shaft. An emergency governor is also furnished which closes the throttle valve in case the pressure at the exhaust nozzle rises above a predetermined point, or in case of an increase of speed of more than 8 per cent. above normal.

Steam is supplied to the turbines by two 6-inch pipes, one port, one starboard, from the main steam line in the forward fireroom. There is a separator in each line and a cross connection between the two lines. There is one condenser on each

side of the dynamo room, the turbines exhausting either to the condensers or to the atmosphere.

An independent air pump takes the water from its own condenser and discharges to a hotwell tank, from which it is pumped to the auxiliary feed-pump suction line. The circulating pump is of the centrifugal type driven by a 6 by 6-inch vertical steam engine.

The dynamo condenser is of the return-flow horizontal type, oval in section, 5 feet 9¼ inches by 4 feet and ½ inch and 6 feet 8½ inches between tube sheets. Each condenser has 2.190 tubes, ⅝-inch outside diameter, No. 16 B.W.G., giving a cooling surface of 2,403 square feet each.

The lighting system comprises a total allotment of about 2,000 lighting fixtures, not including special outlets for signal lanterns, trucks, lights, Ardois signal sets and outlets for special fixtures. There are two mercury-vapor lights in each engine room and in fireroom.

There are sixteen 36-inch searchlights, variously disposed on the superstructure, masts and crane towers. Separate and distinct from the lighting system is the power system, which supplies energy to all auxiliary machinery run by electric motors.

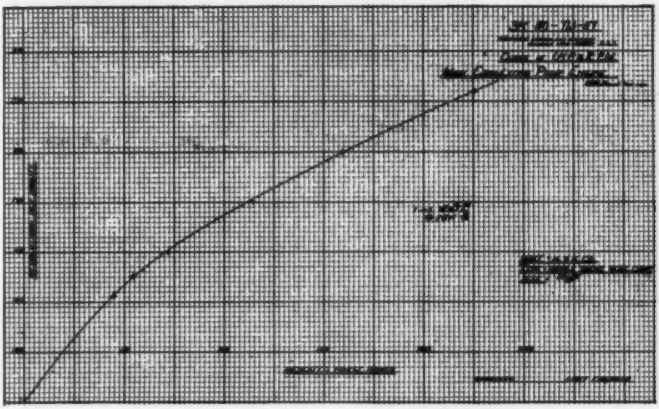
A complete system of wireless telegraphy is installed.

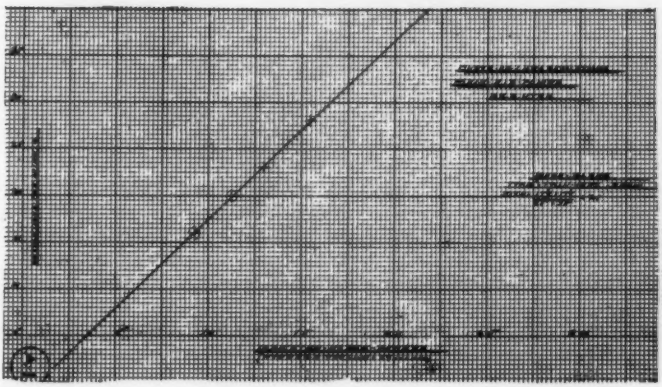
Besides the lighting and power systems there is an interior-communication system, which includes call bells, general alarms, shrill whistle, fire alarms, voice-tube telegraphs, indicators and submarine-signal apparatus.

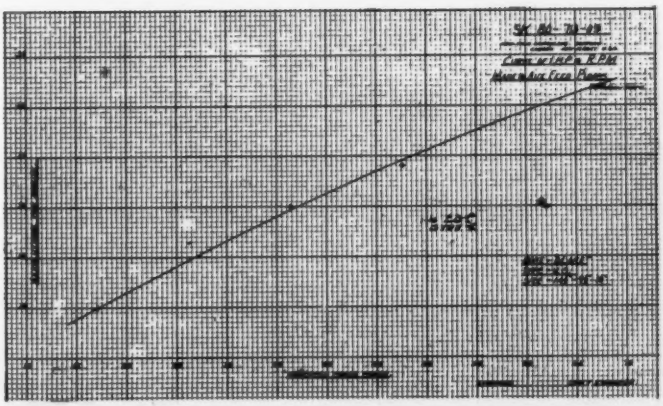
A system of mechanical telegraphs is also installed for communication between the bridge and engine rooms.

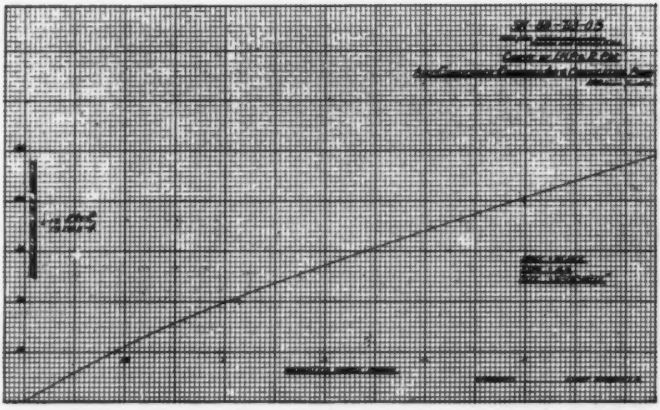
TRIALS OF AUXILIARIES.

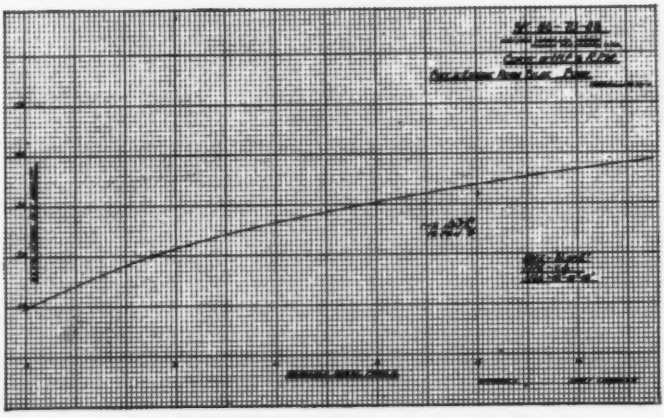
Before the contract trials took place the various auxiliaries were run and indicator cards taken, from which curves of indicated horsepower and strokes or revolutions per minute were plotted. These curves follow on pages 785–787.

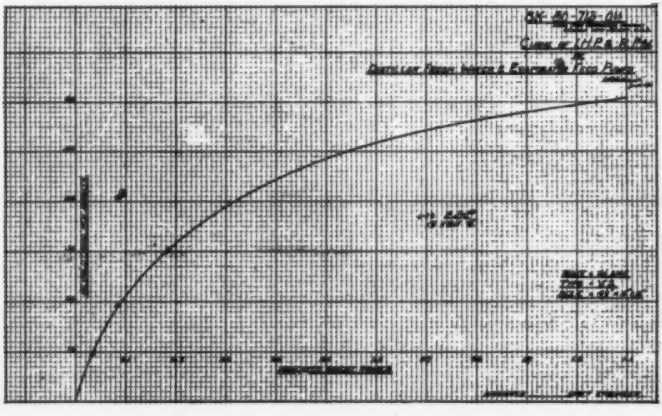


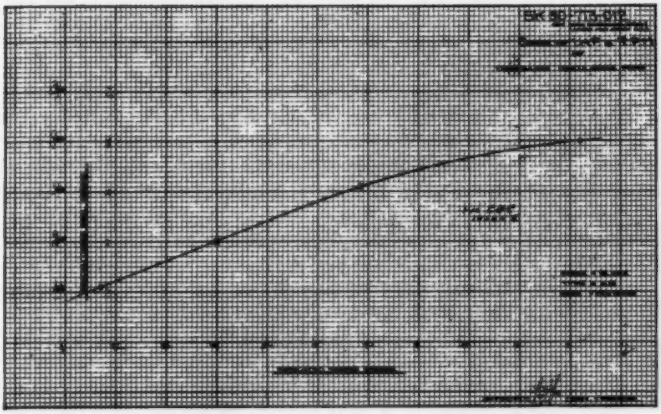


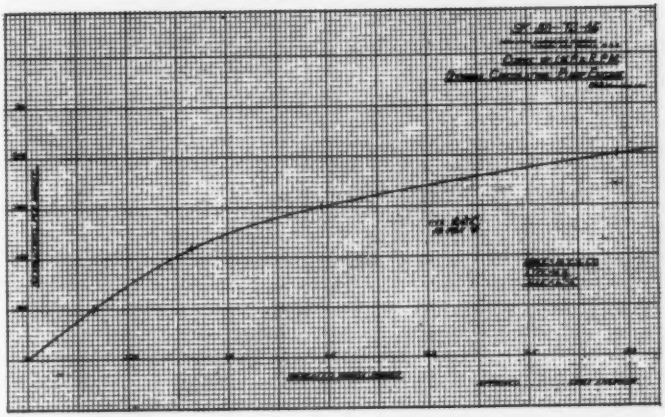


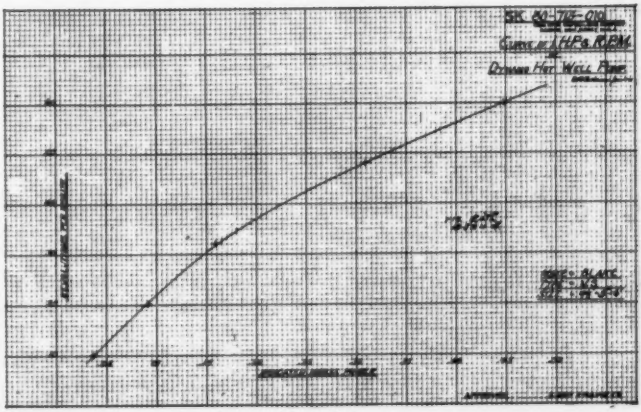






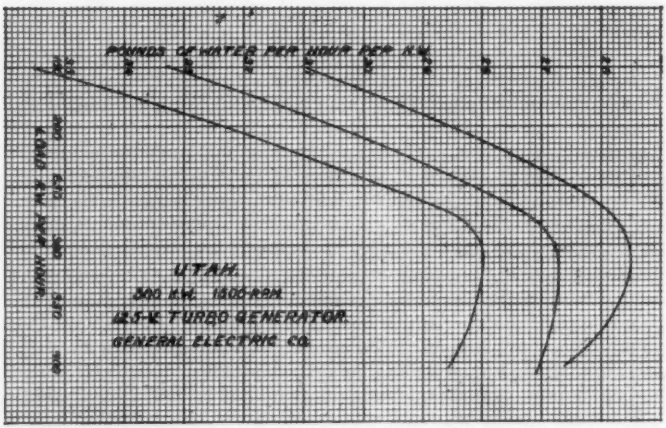






WATER CONSUMPTION OF TURBO-GENERATORS.

Tests for water consumption were made on the turbo-generators at the works of the manufacturers, the General Electric Company, and the results are plotted on the following curves.



PUMPS AND PUMP CONNECTIONS.

No.	Pumps.	Size.	Kind.	Suction pipes from—		Discharge pipes to—		Location.
		inches.		*ins.*		*ins.*		
2	Main air	Two 17½-in., two 35-in. by 21.	Twin vert. bucket, single acting.	12 5	Condenser Turbine drains.	10	Feed tank	One each engine room.
2	Main circulating	11½×23×12	Vert. compound, centrifugal.	2–21 15½	Sea Main drain	30 30	Condenser Overboard.	One each engine room.
4	Main feed	14½×9½×18	Vert. piston, double acting, single.	5 5	Feed-tank connection pipe Channel-way cross-connection pipe.	4	Main feed discharge	Two each engine room.
3	Auxiliary feed	14½×9½×18	....do....	5 3½ 3½ 2½	Feed suction pipe C. and R. pipes from ship's sides (*a*). Reserve feed tanks Hose connections.	4 3 3½	Auxiliary feed main Overboard. Reserve feed tanks (*a*).	One each boiler room.
4	E. R. fire and bilge	12×10×18	....do....	5 5 4½ 5 1½	Sea E. R. bilge Shaft alley C. and R. drainage manifold. Hose connection	5 5 5 3½ 2½	Fire main Overboard. Sanitary. Oil cooler. Hose connection.	Two each engine room.
3	Fireroom fire and bilge	12×10×18	....do....	5 5 5 1½	Fireroom floors Sea C. and R. drainage manifold. Hose connection	5 5 5 2½	Fire main Overboard. Sanitary. Hose connection.	One each boiler room.
2	Oil cooler, circulating	8×7×8	....do....	4	Sea	3½	Oil cooler	One each engine room.
6	Fuel-oil service	5×3×4	Vert. piston, double acting, duplex.	2	Settling tanks	1½	Burners	Two each boiler compartment.
1	Dynamo turbine bearing, circulating.	3½×3½×6	Horizontal piston, double acting, single.	2	Sea	2	Dynamo turbine bearing	Dynamo room.
2	Fuel-oil supply	6×6½×12	Vert. piston, double acting, single.	4	Storage tanks	3½	Settling tank	One each engine room.
4	Forced-lubrication serv.	10×9×12	....do....	4½	Lubricating oil tanks	4½ 3½	To coolers From coolers to bearing.	Two each engine room.
2	Dynamo-room hot well	4½×5×6	....do....	2½	Dynamo-room hot-well tank	2	Auxiliary feed suction in forward fireroom.	Dynamo room.
2	Distiller, circulating	12×14×16	Horizontal piston, double act., single.	6½	Sea	6 5	Distillers and fire main Sanitary system.	Near evaporators.
2	Aux. air and circulating	6×10×10×12	Combined. Horizontal simplex piston.	5 6	Auxiliary condenser Sea	4½ 6	Main feed tanks Overboard.	One each engine room.
2	Dynamo condenser, air	9×18×12	Vert. piston, double acting, single.	7	Dynamo condenser	6	Dynamo-room hot well	Dynamo room.
2	Dynamo condenser, circ	6×6	Centrifugal. Vertical engine.	9	Sea	9	Dynamo condenser	Dynamo room.
2	Evaporator feed	4½×5×6	Vert. piston, light serv., dbl. act., single.	2½ 2	Sea Evaporator feed heaters.	2	Evaporators	Near evaporators.
1	Distiller, fresh water	4½×5×6	Vert. piston, light service.	2½ ...	Bottom of distillers Test tank	... ...	Fresh-water tanks Main feed tanks. Reserve feed-water tanks.	Where directed.
2	Pipe insulator, circulating.	6×8×8	Vert. piston, double acting, single.	3½	Sea	3	Pipe insulators	One each engine room.
3	Ash ejector	12×7×12	Vert. duplex piston	5	Sea	4	Ash ejectors	One each fireroom.

(*a*) No. 2 boiler room only.

DATA.

	Standardization runs.*		24-hour, 12 knots.		24-hour, 19 knots.		4-hour, full power.		2-hour, oil and coal.	
	S.	P.	S.	P.	S.	P.	S.	P.	S.	P.
Miscellaneous:										
Date and time of beginning trial	June 26, 1911.		June 29, 1911, (6:45 P. M.)		July 1, 1911, (1:15 A. M.)		July 2, 1911, (9:40 A. M.)		July 2, 1911, (4:50 P. M.)	
end of trial	.....		June 30, 1911, (6:45 P. M.)		July 2, 1911, (1:15 A. M.)		July 2, 1911, (1:40 P. M.)		July 2, 1911, (6:50 P. M.)	
Draught, forward, feet and inches	27-07½		27-08½		27-08⅛		28-05⅜		27-03⅛	
aft, feet and inches	28-01½		28-01		28-00⅞		28-03⅞		28-04	
Trim by the stern, inches	05½		04⅞		03⅛		09⅛		12⅜	
Displacement, mean, tons	21,268		21,299		21,292		21,284		21,261	
Corresponding draught, feet and inches	27-10½		27-10⅜		27-10¼		27-10⅜		27-09⅛	
Course of ship	.....		Various		Various		Various		Various	
Wind	.....		Calm, fog		Light breeze		Light breeze		Light breeze	
Sea	.....		Smooth		Smooth		Smooth		Smooth	
Number of boilers in use	.....		6		12		12		12	
Speed of ship, knots per hour	.....		12.018		19.225		21.042		21.04	
Slip of propeller, in per cent. of its own speed	.....		18.27		18.54		22.25		28.54	
Torsion-meter readings, in millimeters, average, No. 1	.....		15.5		56.67		88.61		90.13	
No. 2	.....		31.21		89.06		91.328		89.88	
No. 3	.....		31.87		57.21		86.672		85.73	
No. 4	.....		15.54		52.28		88.469		89.29	
constant, No. 1	.....		.....		.....		0.23269		.....	
No. 2	.....		.....		.....		0.23733		.....	
No. 3	.....		.....		.....		0.23382		.....	
No. 4	.....		.....		.....		0.23753		.....	
Voltage at switch board	.....		125		125		125		125	
Total output of generators, ampères	.....		904.1		1,740		2,039		2,586	
Power output for F. D. blowers, ampères	.....		.....		842		1,144		1,663	
Pressure:										
Main steam, at boilers, pounds, maximum	.....		215		223		225		222	
mean	.....		199		208		213		205	
engine rooms, pounds, maximum	205	207	215	207	210	214	200	201	190	193
mean	...	...	199	199	198	206	195.6	195	185	189
H.P.C. turbines, pounds, mean	...	121	...	114.9	Idle		Idle		Idle	
I.P.C. turbines, pounds, mean	180	...	46.9	...	177	...	Idle		Idle	
M.H.P. turbines, pounds, mean	...	...	20 9	20.9	81	81	140.7	139	141.6	141
maximum	149	149	22	22	85	85	145	141	145	148
L.P. turbines, pounds, mean	15.5	14	3.09	3.36	8.5	9.1	14.94	14	14.9	12.75
Vacuum in condensers, inches of mercury, maximum	29.5	29.3	29.4	29.5	28.7	28.8	28.2	28 2	28.1	28.2
mean	...	...	29 13	29.15	28.5	28.6	28.09	28.2	28	28
Main feed line, at pumps	...	...	275		270		260		265	
between heaters and boilers	...	...	235		230		220		225	
Auxiliary exhaust in heater, pounds, gage	7	8	4.1	6.2	5	6	4.28	5.75	2.5	4.8
Forced-lubrication system, pounds, gage	15	15	13.5	13.35	13	12	12.9	12	12	11.9

* Mean of five high runs.

Pressure:										
Fuel-oil line, fireroom No. 1	.....		.....		.....		.....		208	
No. 2	.....		.....		.....		.....		208	

| | | |

	Standardization runs.*		24-hour, 12 knots.		24-hour, 19 knots.		4-hour, full power.		2-hour, oil and coal.	
	S.	P.	S.	P.	S.	P.	S.	P.	S.	P.
Horsepower:										
No. 1 fireroom fuel-oil service pumps	.....		.....		.....		.....		20	
No. 2 fireroom fuel-oil service pumps	.....		.....		.....		.....		20	
No. 3 fireroom fuel-oil service pumps	.....		.....		.....		.....		20	
I.H.P. chargeable to F.D. blowers	.....		.....		162		220		245	
of ice machines	.....		20		20		.....		.....	
Other auxiliaries	.....		186		184		186		182	
Total I.H.P. of all engineer's auxiliaries	.....		542		838		1,100		1,097	
Shaft H.P. of all turbines	28,842		2,033	1,905	10,128	7,022	13,742	13,294	13,688	13,366
Total	.....		4,481		17,988		28,136		28,151	
Effective H.P. for bare hull	.....		2,200		9,750		15,903		15,903	
Water consumption, pounds (Steam, pounds per hour):										
For all turbines and engines' auxiliaries	.....		99,498		262,498		406,397		.....	
Per S.H.P. main turbines only	.....		25.0122		15.3001		15.0317		.....	
H.P. of main turbines and engineer's auxiliaries	.....		21.98		14.587		15.032		.....	
lighting and ventilation, estimated	.....		3,356		5,959		6,961		.....	
domestic purposes, estimated	.....		100		100		100		.....	
steering engine, estimated	.....		600		600		600		.....	
evaporating plant, estimated	.....		4,764		9,528		.....		.....	
Coal used on trial:										
Quality and kind	.....		George's Cr., good		Pocahontas, good		Pocahontas, good		Pocahontas, good	
Heat value, B.T.U. per pound	.....		14,263		14,874		14,717		14,795	
Total burned, pounds	.....		266,040		633,432		174,048		62,048	
per square foot grate surface, pounds	.....		372.6		443.568		121.884		43 45	
S.H.P. of main turbines, pounds	.....		67.560		36.936		6.436		2.294	
total H.P., pounds	.....		59.376		35.208		6,184		2.204	
Water evaporated per pound of coal, pounds	.....		8,886		9.942		9.34		.....	
Coal burned per hour, pounds	.....		11,085		26,393		43,512		31,024	
Knots per ton of coal	.....		2.429		1.636		1.083		1.519	
Fuel oil:										
Specific gravity at 83 degrees F	.....		.....		.....		.....		0.8859	
Heat value B.T.U. per pound	.....		.....		.....		.....		19,421	
Total burned, pounds per hour	.....		.....		.....		.....		8,617.3	
per square foot grate, pounds per hour	.....		.....		.....		.....		6.03	
Coal per hour equivalent to oil burned	.....		.....		.....		.....		11,312	
Total coal per hour and coal equivalent of oil	.....		.....		.....		.....		42,336	
Pounds of coal per pound of oil	.....		.....		.....		.....		3.6	
Deduced data:										
S.H.P. total per square foot of grate surface	.....		5.516		12.01		18.93		18.945	
Pounds of coal per hour per S.H.P. main engines	.....		2.815		1.539		1.609		1.147	
all machinery	.....		2.474		1.467		1.546		1.102	
square foot grate surface	.....		15.525		18 482		30.471		21.725	
heating surface	.....		0.3451		0 4108		0.6775		0.4829	
Cooling surface (main condenser), sq. ft. S.H.P. (turbines)	.....		7.737		1.777		1.127		1.126	
Heating surface, square feet per S.H.P. (total)	.....		8.155		3.746		2.376		2.376	

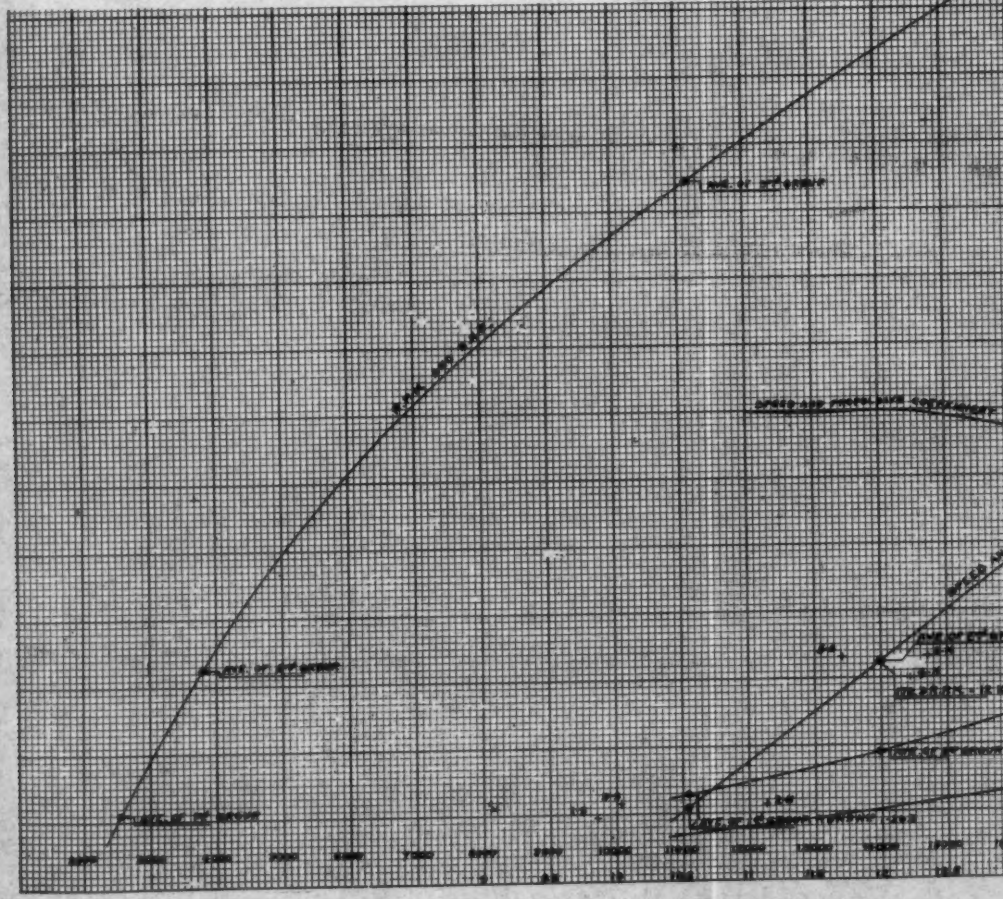
Heating surface, square feet per S.H.P. (total)	.....	2.155	3.746	2.376	2.376

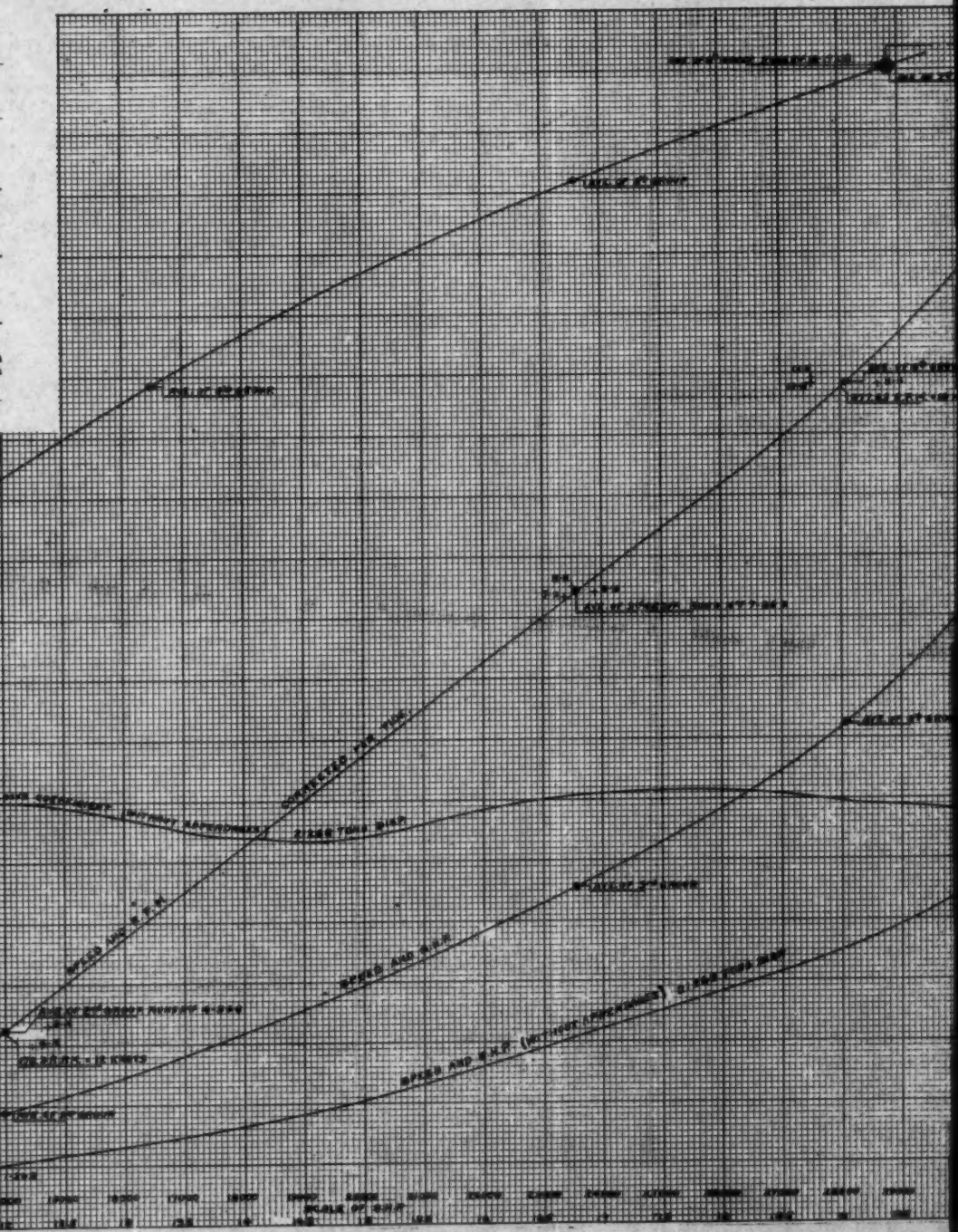
U. S. S. "UTAH."

Standardization Trial Over the Measured-Mile Course, Rockland, Maine, June 25, 1911.

Speed.	Functions for averages of speed.	R.P.M.	Functions for averages of R.P.M.	S.H.P.	Functions for averages of S.H.P.
9.874	×1= 9.874	149.46	×1=149.46	2,458	×1= 2,458
11.153	×2=22.306	151.32	×2=302.64	2,633	×2= 5,266
10.045	×1=10.045	151.70	×1=151.70	2,657	×1= 2,657
	4)42.225		4)603.80		4)10,381
	10.556 av.		150.95 av.		2,595 av.
12.035	×1=12.035	170.02	×1=170.02	3,695	×1= 3,695
11.728	×2=23.456	173.10	×2=346.20	3,851	×2= 7,702
12.367	×1=12.367	173.25	×1=173.25	3,893	×1= 3,893
	4)48.058		4)689.47		4)15,290
	12.015 av.		172.37 av.		3,823 av.
16.696	×1=16.696	243.40	×1=243.40	10,933	×1=10,933
16.771	×2=33.542	245.07	×2=490.14	11,254	×2=22,508
16.958	×1=16.958	244.20	×1=244.20	11,157	×1=11,157
	4)67.196		4)977.74		4)44,598
	16.799 av.		244.44 av.		11,150 av.
18.765	×1=18.765	278.23	×1=278.23	16,422	×1=16,442
19.329	×2=38.658	278.10	×2=556.20	16,470	×2=32,940
18.774	×1=18.774	279.08	×1=279.08	16,562	×1=16,562
	4)76.197		4)1,113.51		4)65,944
	19.049 av.		278.38 av.		16,486 av.

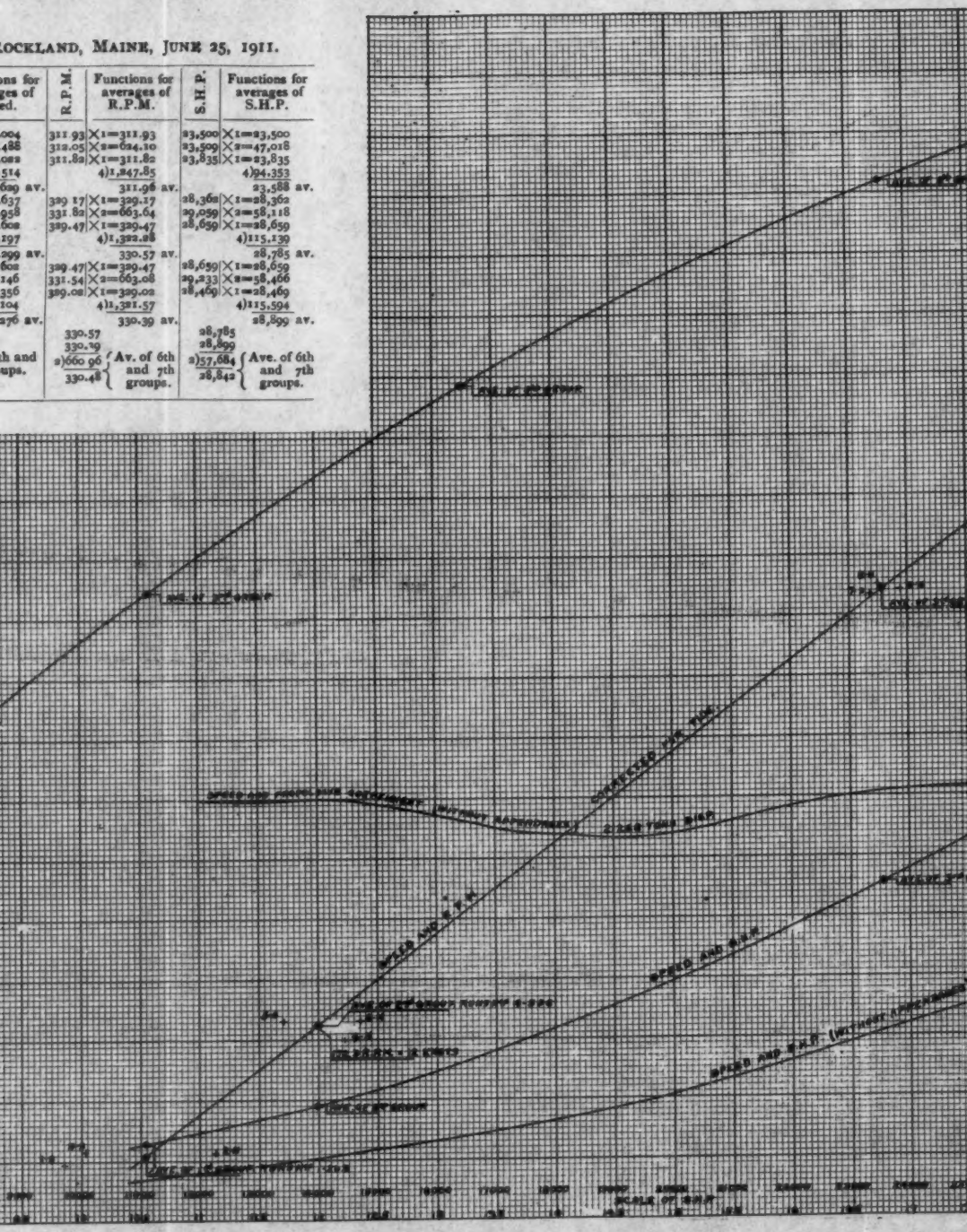
No. of run.	Speed.	Functions for averages of speed.	R.P.M.	Functions for averages of R.P.M.	S.H.P.	Functions for averages of S.H.P.
13-S	21.004	×1=21.004	311.93	×1=311.93	23,500	×1=23,500
14-N	20.244	×2=40.488	312.05	×2=624.10	23,509	×2=47,018
15-S	21.022	×1=21.022	311.82	×1=311.82	23,835	×1=23,835
		4)82.514		4)1,247.85		4)94,353
		20.629 av.		311.96 av.		23,588 av.
16-S	21.637	×1=21.637	329.17	×1=329.17	28,362	×1=28,362
17-N	20.979	×2=41.958	331.82	×2=663.64	29,059	×2=58,118
18-S	21.602	×1=21.602	329.47	×1=329.47	28,659	×1=28,659
		4)85.197		4)1,322.98		4)115,130
		21.299 av.		330.57 av.		28,785 av.
18-S	21.602	×1=21.602	329.47	×1=329.47	28,659	×1=28,659
19-N	21.073	×2=42.146	331.54	×2=663.08	29,233	×2=58,466
20-S	21.356	×1=21.356	329.02	×1=329.02	28,469	×1=28,469
		4)85.104		4)1,321.57		4)115,594
		21.276 av.		330.39 av.		28,899 av.
	21.299		330.57		28,785	
	21.276		330.39		28,899	
	2)42.575	Ave. of 6th and 7th groups.	2)660.96	Av. of 6th and 7th groups.	2)57,684	Ave. of 6th and 7th groups.
	21.288		330.48		28,842	





ROCKLAND, MAINE, JUNE 25, 1911.

ons for ges of ed.	R.P.M.	Functions for averages of R.P.M.	S.H.P.	Functions for averages of S.H.P.
004	311 93	×1=311.93	23,500	×1=23,500
488	312.05	×2=624.10	23,509	×2=47,018
082	311.82	×1=311.82	23,835	×1=23,835
514		4)1,247.85		4)94,353
629 av.		311.96 av.		23,588 av.
637	329 17	×1=329.17	28,362	×1=28,362
958	331.82	×2=663.64	29,059	×2=58,118
602	329.47	×1=329.47	28,659	×1=28,659
197		4)1,322.28		4)115,139
299 av.		330.57 av.		28,785 av.
602	329.47	×1=329.47	28,659	×1=28,659
146	331.54	×2=663.08	29,233	×2=58,466
356	329.02	×1=329.02	28,469	×1=28,469
104		4)1,321.57		4)115,594
276 av.		330.39 av.		28,899 av.
	330.57		28,785	
	330.39		28,899	
th and ups.	2)660 96	Av. of 6th and 7th groups.	2)57,684	Ave. of 6th and 7th groups.
	330.48		28,842	



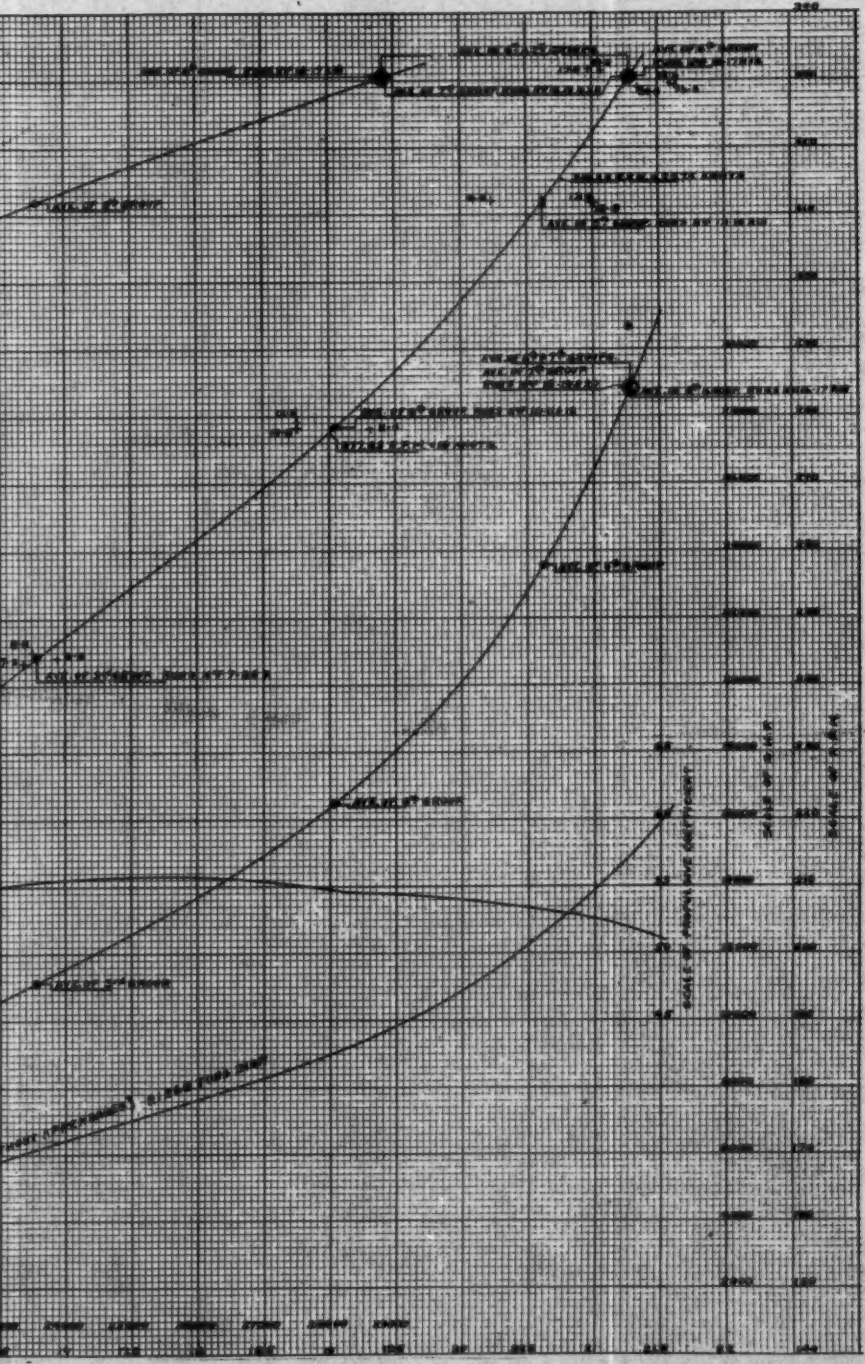
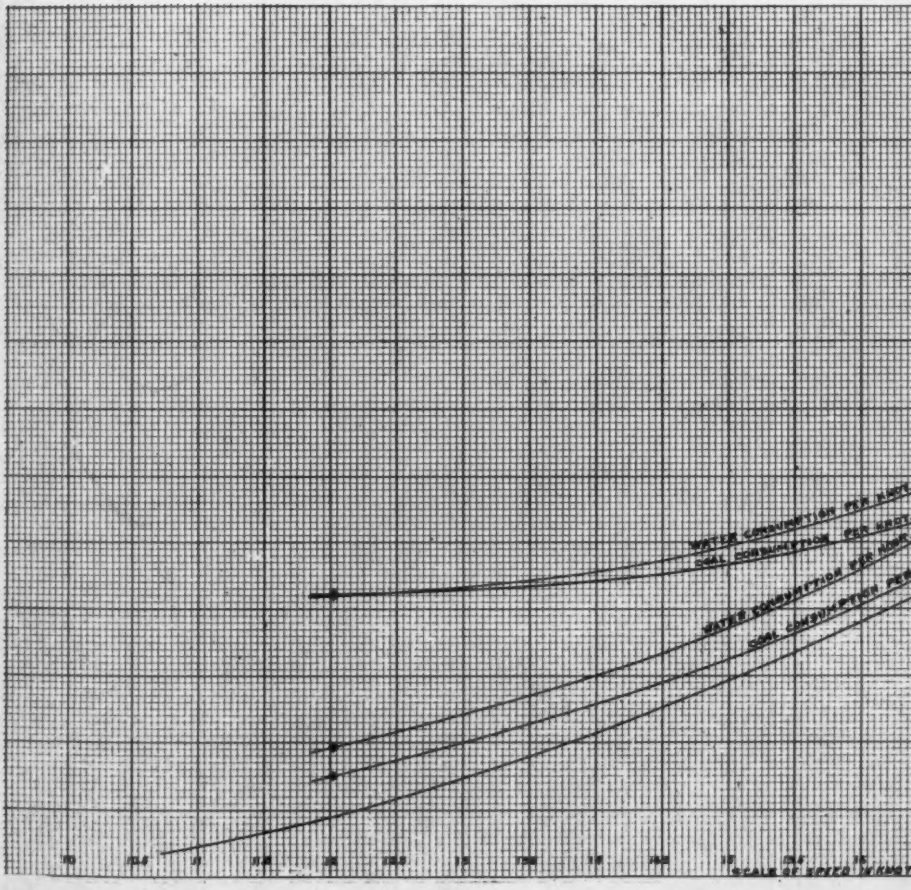


Plate XI.

U. S. S. "UTAH."

PRELIMINARY TRIALS OFF ATLANTIC COAST, JUNE 29 AND 30, JULY 1 AND 2, 1911.



1911.

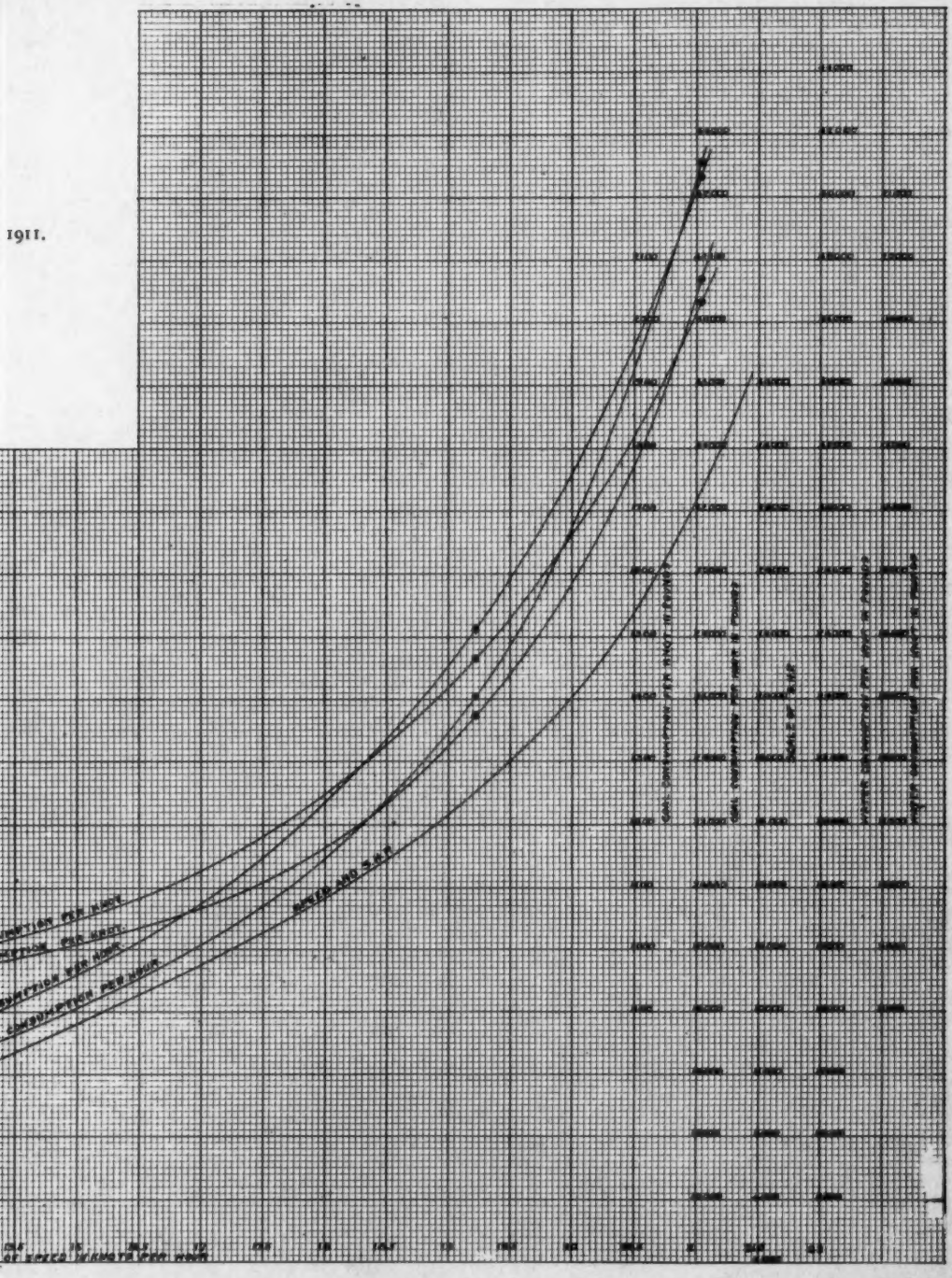


Plate XII.

STANDARDIZATION TRIALS.

These trials took place on the Rockland, Maine, course, June 26th, 1911. The results are given in the table and plotted in the curves on Plate XI.

FULL-POWER AND ENDURANCE TRIALS.

Owing to prolonged fog conditions these were delayed, but were begun on June 29th and concluded July 2d, 1911. The resulting data is given in the table on pages 789–791.

The water consumptions are plotted on the curves on Plate XII.

POST-TRIAL INSPECTION OF MACHINERY.

The following is a summary of the report of the inspection of the machinery after the trial:

Boilers.—These were all opened and carefully examined. The tubes were found to be all straight and in good condition as regards fire action, corrosion, etc. There was considerable mud in the tubes. All main and auxiliary-feed checks were tested, and in most instances were found to be tight. Where they were not tight they were reground and made so. Generally speaking, all parts of the boilers are in excellent condition.

Turbine Casings.—The following turbines were examined, with results as indicated.

High-Pressure Cruising.—Condition good. Slight rust in places from vapor.

Intermediate-Pressure Cruising.—Condition good. Slight rust in places from vapor.

Starboard Main High Pressure.—Rotor blades in first expansion showed signs of having rubbed on lower part of casing. A few blades in lower casing also showed signs of having rubbed on the rotor. .003-inch was taken off such rotor and casing blades as showed signs of having rubbed, the rotor being lifted to permit of complete examination of all blades and

the increase of clearance as stated. Slight rust in places from vapor.

Starboard Low Pressure (and Backing).—Good condition. Considerable rust, particularly on upper 30 degrees of the top casing.

Port Main High Pressure.—This turbine was opened on account of the lack of clearance found on the other main high pressure. Condition good. Very little rust.

Pumps.—One Main Feed Pump.—Condition good.

One Main Air Pump.—Condition good.

One Fire and Bilge Pump in Engine Room.—Condition good.

One Fire and Bilge Pump in Fire Room.—Condition good.

One Auxiliary-Feed Pump.—Condition good.

One Oil Pump, each kind (Starboard forced-lubrication, outboard)—Condition good.

One Dynamo Air Pump.—Condition good.

One Evaporator-Feed Pump.—Condition good.

One Distiller Circulating Pump.—Condition good.

Fill and Test All Condensers (Main augmenter, auxiliary and dynamo).—Found to be tight under weight of water. No additional pressure applied.

One Main Circulating Pump Engine.—Condition good.

NOTES ON OIL FUEL AND ITS COMBUSTION.

COMPILED BY LIEUTENANT A. T. CHURCH, U. S. NAVY.

School of Marine Engineering.

The liquid fuels commonly known are (1) crude coal tar, (2) tar and creosote oil, (3) shale oil, (4) crude petroleum, and (5) petroleum residue or refuse. The first three cannot be considered for Navy use on account of insufficient supply, although they are used to a slight extent in the merchant marine and in foreign navies.

Crude coal tar is formed during the destructive distillation of coal for the production of coal gas. It contains in its crude condition so much solid matter that it can only be employed with very unsatisfactory furnace arrangements.

Tar oil is obtained when tar is distilled. First the light oils pass over, from which benzine and its homologues can be obtained; then heavy tar oil, from which carbolic acid is made; then pitch, from which anthracene is separated. The heavy tar oil obtained in this way from coal tar is often called creosote, and forms a fairly good liquid fuel. Its specific gravity is greater than that of water, being about 1.1, and it is known as heavy oil. The oil obtained from blast furnaces is sometimes used locally for fuel under the name of phenoloid. Both creosote and phenoloid have the great disadvantage that in closed firerooms they give off fumes, probably of acrolin, which painfully effects the eyes.

Shale oil comes mostly from Scotland, and is obtained from the shales. The shales differ from coal in that they are a mixture of hydrocarbons and mineral matter instead of carbon with hydrocarbons. When heated to redness in a retort they

do not form coke, but leave 70 or 80 per cent. of mineral matter. The oils pass over at about 900 degrees F., and when refined yield about the following:

Naphtha	per cent. 6
Burning oil	32
Heavy oil	24
Paraffin scale	12
Loss	26

The heavy oil is used as fuel. It is heavier than water, has a high flashing point, and makes an excellent fuel.

Petroleum.—There are three kinds of petroleum is use, namely, those which on distillation yield (1) paraffin, (2) asphalt, (3) olefin. To the first group belong the oils of the Appalachian range and middle west. They are dark brown with a greenish tinge. Upon distillation they yield such a variety of light oils that their value is too great to permit their general use as fuels.

To the second group belong the oils from Texas and California. These vary from reddish brown to jet black, and are used mostly for fuel.

The third group comprises oils from Russia, which are also used more extensively for fuel than for any other purpose.

Crude petroleum as it comes from the well is a mixture of many hydrocarbons and varies a great deal both in its chemical and physical properties according to its locality. Crude petroleum is subjected to fractional distillation before use, and only the heavier portions are used for fuel. Some of the western oils, the heavier ones, and some other oils, such as Borneo oil, can be burned in the crude state, but the greater part have to be refined. The crude oil contains sediment and other impurities, such as sulphur, which clog the burners, corrode them, and reduce the efficiency of the oil as a fuel. The lighter volatiles of crude oil are highly inflammable and distill over at a low temperature, which makes the crude product dangerous.

REFINING.

The crude oil is distilled in iron retorts heated by a fire, oil burners usually being employed. The retort is simply a large tank, shaped like a boiler drum, with furnace underneath. It is fitted with oil-supply and vapor pipes with necessary valves, and also safety valves set at about 5 pounds. The products of distillation are passed through condensers which consist of straight lengths of pipe connected by return bends and contained in a large vessel of water. The distillate is turned into separate vessels as the temperature rises, so that four distinct distillates are usually obtained, light and heavy naphthas and light and heavy oils, the division between the four varying in different refineries. Pennsylvania petroleum would yield: Naphtha 8 per cent. to 20 per cent., oils 76 per cent. to 78 per cent., residue 5 per cent. to 9 per cent., and loss about 5 per cent. The residue is used for fuel oil. A Pennsylvania and Russian oil would yield about the following:

Pennsylvania.

Gasoline	per cent.	1.5
Petroleum spirit, C. Naphtha		10.0
Petroleum spirit, B. Naphtha		2.5
Petroleum spirit, A. Naphtha		2.5
Lamp oil		54.0
Lubricating oil		17.5
Paraffin solid		2.0
Residue		6.0
Loss		10.0

Russian.

Petroleum spirit	per cent.	6.0
Lamp oil		20.0
Solar oil		6.0
Astatki (liquid fuel)		58.0

The percentage of fuel oil obtained from the crude in our western fields is from 40 to 60 per cent.

OILS FROM VARIOUS LOCALITIES COMPARED.

The elements found in the different oils are in about the following proportions:

Pennsylvania Crude.

Carbon	per cent.	84.9
Hydrogen		13.7
Oxygen		1.4

Caucasian Light Crude.

Carbon	per cent.	86.2
Hydrogen		13.7
Oxygen		.1

Caucasian Heavy Crude.

Carbon	per cent.	86.6
Hydrogen		12.3
Oxygen		1.1

Petroleum Refuse (Russian).

Carbon	per cent.	87.1
Hydrogen		11.7
Oxygen		1.2

In the better grades of our western oils, treated at the wells before shipment, in which moisture has largely been eliminated, it can be roughly assumed that 3 per cent. of the oil is made up of sulphur, nitrogen, oxygen and water. This relationship is not universal, certain southern California oils containing large percentages of sulphur.

The predominating oil used on the Pacific Coast, known as Bakersfield oil, averages about 16 degrees Baumé. This makes a 42-gallon barrel weigh about 336 pounds. The ultimate analysis of this oil is:

Carbon .... per cent.	85.0	Nitrogen ... per cent.	.2
Hydrogen. .	13.0	Oxygen.. ..	1.0
Sulphur ...	.8	Water	1.0

A number of lighter oils in general use ranging in the neighborhood of 18 to 20 degrees Baumé would average about as follows:

Carbon .... per cent.	84.0	Nitrogen ...	.2
Hydrogen ..	13.0	Oxygen ...	1.0
Sulphur ...	.8	Water	1.0

Certain heavier oils ranging from 12 to 14 degrees Baumé give—

Carbon .... per cent.	86.0	Nitrogen ..	.2
Hydrogen ..	11.0	Oxygen ...	1.0
Sulphur ...	.8	Water	1.0

An analysis of Texas oil (Beaumont crude), used by the Navy Fuel Board, gave—

Before Distillation.

Carbon	per cent.	84.6
Hydrogen		10.9
Sulphur		1.63
Oxygen		2.87

After Distillation Atmospheric Pressure to 680 degrees F.

Carbon	per cent.	83.26
Hydrogen		12.41
Sulphur		.50
Oxygen		3.83

The amount of sulphur in this oil varies from 2 to 3 per cent.

In Texas oil refineries the crude oil is forced through tubes of a condenser-like apparatus, the hot residue from each still flowing around the tubes. This residue may be fuel oil or asphalt. The crude now goes into the first of a series of about eight stills, and then in succession to the other stills, finally being drawn out around the tubes of receptacle first mentioned, thus serving to heat up the crude going in, and is in turn

cooled by it. The temperature of each still is higher than the one preceding it, and each still is connected to a vertical condenser, where products formed at different temperatures are obtained. Oil burners are used to heat the stills.

Supply.—The following table gives the petroleum-producing countries and the production from 1905–1909, inclusive:

Barrels of 42 gallons.

.....	1905.	1906.	1907.	1908.	1909.	Rank.	Per cent.
United States.....	134,717,580	126,493,936	166,095,335	178,527,355	188,134,274	1	61.24
Russia.....	54,960,270	58,897,311	61,850,734	63,186,447	65,970,350	2	22.19
Galicia.....	5,765,317	5,467,967	8,455,841	12,612,295	14,932,799	3	5.02
Dutch E. Indies.	7,849,896	8,180,657	9,982,597	10,283,357	11,041,852	4	3.71
Roumania	4,420,987	6,378,184	8,118,207	8,252,157	9,321,138	5	3.13
India.....	4,137,098	4,015,803	4,344,162	5,047,038	6,676,517	6	2.24
Mexico.	...	...	1,000,000	3,481,410	2,488,742	7	.84
Japan	1,472,804	1,710,768	2,010,639	2,070,929	2,012,409	8	.68
Peru	447,880	536,294	756,226	1,011,180	1,316,118	9	.44
Germany.....	560,963	578,610	756,631	1,009,278	1,018,837	10	.34
Canada	634,095	569,753	788,872	527,987	420,755	11	.14
Italy	44,027	53,577	59,875	50,966	50,000	12	.03
Other.....	30,000	30,000	30,000	30,000	30,000	...	...
	215,040,917	212,912,860	264,249,119	285,090,399	287,413,791		

In 1910 the increased production in the United States came to between 200,000,000 and 204,000,000 barrels, which was more than the whole world produced seven years ago, and represents two-thirds of the world's supply today. California, owing to the remarkable developments during the year in the Sunset Midway district, leads, with 73,000,000 barrels; the Mid-Continent and Rock Mountain fields follow, with 53,000,000, while Illinois and the Appalachian and Lima, Indiana, fields follow with 32,000,000 each, and the Gulf and Caddo fields furnish 14,000,000.

Oil depots are now established at the following places: London, Liverpool, Hamburg, Antwerp, Rouen, St. Louis (France), Enna, Constanza, Noworowik (Batun), Alexandria, Suez, Sorenbar, Mombasa, Colombo, Madras, Calcutta, Rangoon, Penang, Batavia, Surabaya, Singapore, Bangkok, Saigon, Hongkong, Swatow, Foochow, Amoy, Hang-

kow, Shangkai, Yokohama, Kobe, Nagasaki, Sidney, Melborne, Adelaide.

Calorific Value.—A good sample of fuel on analysis gives C = 87.80 per cent.; H = 10.78 per cent.; and O = 1.24 per cent., which gives E = 19.9 pounds of water evaporated per pound of fuel. Taking 965.8 B.T.U.'s as the heat required to convert one pound of water at a temperature of 212 degrees F. into steam at the same temperature, there are 965.8 × 19.9 = 19,219 B.T.U.'s in a pound of fuel. If average steaming coal evaporates about 14.66 pounds of water per pound of coal theoretically, the theoretical value of oil would be 1.36 times that of coal.

A paper published by A. C. Sherman and A. H. Kropff in the Journal of the American Chemical Society, in October, 1908, gives the calorific values of a number of representative American petroleum oils, and also calculated values by using the following formula: B.T.U. = 18,650 + 40 (Baumé —10).

Their object was to point out an approximate relationship between the density and the calorific power. Some of the results are tabulated.

	B.T.U. per pound.	B.T.U. calculated by above formula.
Pennsylvania fuel oil	19,656	19,526
Kansas crude	19,389	19,578
Texas crude	19,242	19,332
Indian Territory	19,418	19,342
California crude	18,799	19,150
California refined	19,555	19,530

A summary of these tests shows that sixty-four samples of petroleum oils, ranging from heavy crude to gasoline, and representing the products of the principal oil fields of the United States, were examined for calorific power by combustion in oxygen in the Atwater-Mahler bomb calorimeter with results ranging from 18,572 to 21,120 British thermal units per pound.

In general the decrease in calorific power with increase in

gravity was fairly regular, so that the relation between the two may be expressed approximately by means of a simple formula. When the calorific power calculated from the densities by means of this formula was compared with those determined by experiment, it was found that in one-ninth of the cases the difference was greater, and in eight-ninths it was less than 1 per cent.; in only one-thirtieth was it greater than 2 per cent.; in no case was it as great as 3 per cent.

While it is obviously improbable that an exact quantitative relation should exist, it is believed that from the data the calorific power of commercially-pure petroleum oils may be predicted from the density, with a sufficient approach to accuracy for practical purposes.

MISCELLANEOUS TESTS.

Besides examining the oil to see that it is not cloudy in appearance from the presence of glue or suspended matter of any kind, the usual tests are for specific gravity, flash point, firing point, viscosity, calorimetry, sediment, water and, sometimes, sulphur.

Specific gravity is determined by means of Baumé hydrometer. A hydrometer jar is about four-fifths filled with the oil, and the depth read off to which the instrument sinks in the oil. This may be effected by placing a strip of white paper back of the jar and noting the point at which the lower meniscus of the oil touches the scale. The temperature of the oil is taken at the same time, and, in case it is not 60 degrees F. (15.5 degrees C.), for every increase of 10 degrees F. (5.5 degrees C.) subtract 1 degree Baumé from the reading of the hydrometer. The approximate specific gravity may be found by the formula $140 \div (140 + d)$, d representing the reading Baumé at 15.5 degrees C.

Oil low in gravity requires an excess of steam for preheating in the auxiliary suction tanks, both for the preliminary treatment of the oil in an attempt to remove the excess of moisture and silt by warming, and also to enable the oil-

pressure pump to lift this oil under suction without danger of interruption in the pumping of oil to the burners. Also the heavier the oil the greater the amount of heating necessary in the pressure heaters between pumps and burners, where heat is added to facilitate atomization of the oil to economize in the amount of steam necessary for such atomization.

For Navy use the oil must have a gravity Baumé not greater than 30 degrees at 60 degrees F.

Flash Tests.—The flash point is the temperature to which an oil must be heated to give off vapors which, when mixed with air, produce an explosive mixture. The test may be conducted by placing oil in an open vessel, heating it, and then passing a light across the top until there is a flash. The appearance of a slight bluish flame shows that the flash point has been reached. This test is rather inaccurate, as any cause producing the rapid evolution of a large amount of petroleum vapor tends to lower the flash point. The faster the oil is heated the lower the flash point, as more vapor is driven out. The liquid evaporates faster from a large and shallow cup; best results are obtained from a deep cup about half filled. The nearer or larger the testing flame the lower the flash point. The point of application of the flame also affects the flash point, the point being too high when the flame is first applied at center of cup; the best point of application is at the edge, the mixture of air and vapor being more complete at this point. Draw the flame from edge diametrically across top of cup. A simple and better method is to use a beaker. Cover it with a disc of asbestos through which a thermometer passes to within a quarter of an inch from the bottom of the beaker. The beaker, etc., is now put in the sand bath, and surrounded with sand to the level of the liquid. A small flame is then applied under the bath, and the temperature allowed to rise about two degrees a minute. After each rise of one degree, the asbestos disc is turned to one side, and a small flame is quickly put into the vapor.

Closed or covered testers are much preferable to the above and are used for tests in the Navy. In these instruments a

metal cup of standard size is used for holding the oil, means being provided for obtaining correct quantity of fuel. This metal cup rests in an outer vessel which forms a water jacket, and which is heated by a lamp placed below. The oil cup is provided with a close-fitting cover which carries a test flame and thermometer. Holes are provided in the cover which can be closed or opened by a sliding piece of metal. When flame is placed below outer water jacket the temperature rises slowly. At regular intervals the slide on the cover is pulled aside, test flame is lowered into the opening in the cover, and the moment a flash is seen to occur the temperature is noted, the point being regarded as the flash point.

If the flash point is comparatively high the fuel oil becomes thick and sluggish, can be burned with difficulty, and great power is required to atomize it before ignition at the furnace. The volatile elements which are not present with a high flash point are a good assistance to combustion within ordinary limits. The extra refining necessary to produce a high flash point also makes the fuel more expensive.

The British Admiralty and our Navy specify a flash point of not less than 200 degrees F. The merchant marine use oil with a flash point between 150 degrees F. and 160 degrees F. No trouble has been experienced in the safe storage and burning of oils with a flash point of 150 degrees F. and upwards.

Firing Point.—The firing point of an oil is the temperature at which it will give off vapor which when ignited will burn continuously. It is made by continuing to heat the oil (the cover being removed in the case of a closed tester without slipping out the thermometer) at the same rate after the flash test is made and noting the point as indicated above. The flame may be extinguished by a piece of asbestos board and the heating discontinued. The firing point is usually about 50 degrees F. above the flash point.

Viscocity Test.—The viscosity test determines the rate of flow of an oil through a specific orifice under a certain head. The Navy specification requires oil to flow freely in

a continuous stream through a ½-inch circular hole under a two-foot head at a temperature of 40 degrees F.

Heating Value.—The value is determined by using the bomb calorimeter. Oil for the Navy must have a value not under 144,000 B.T.U. per gallon. The bomb calorimeter is used to determine the B.T.U. per pound, from which the B.T.U. per gallon can be calculated by using 8,331 pounds of distilled water per gallon at 60 degrees F. and the specific gravity of the oil as determined by the Baumé test at the same temperature.

Sediment.—This is determined by placing 50 c.c. of oil and 50 c.c. of naphtha in a beaker, shaking them, and allowing them to stand for six hours. Percentage of sediment must not be over one per cent.

Water.—Water is determined as in test for sediment. By rubbing the oil together with a little eosin on a glass plate the oil will take on a pink color if water is present. Also with oils which flash below 240 degrees C. in the Pensky apparatus, the moisture test is conducted by determining the losses in weight suffered by two samples of about the same mass (10 to 15 grammes) of the original and the dehydrated oil, when all froth formation has ceased. The difference in the losses of weight of the two samples gives the percentage of water in the original oil. The oil is dehydrated before this heating by being shaken with calcium chloride in a flask and subsequent filtration through a dry filter.

Sulphur.—This may be determined by burning the oil and having the products of combustion sucked through wash bottles containing bromine water. The sulphuric acid is then determined with barium chloride. The odor of sulphur fumes is also noticeable when burning oil containing sulphur. Sulphur in oil should not exceed ¾ of one per cent. by weight.

Storage.—Refined petroleum and fuel oil have very pronounced searching qualities, and its fumes have a great penetrating power, consequently the joint which is quite tight under water pressure may be subjected to leakage with this class of petroleum. As the heavier oil has a high flash point

there is little danger of storing it, and very little trouble; a joint which is watertight is usually tight under the same head pressure of heavy oil. A lower flash point allows some of the more penetrating oils to remain in the fuel, and some trouble might be expected; but when the fuel is stored in the double-bottom compartments the lower temperature of the sea water which is in contact with a large part of the plating surface should prevent any excessive formation of vapor and gases, even from the lower flash-point fuels.

The double bottoms are suitable for storage with little alteration, if tested to a sufficient head. Lloyds specify a head of at least 12 feet above the water line, or the highest point of the tank. They also specify open-topped ventilating pipes to each tank to discharge above the upper deck. Holes should be provided at the highest point of each subdivision of the double bottoms, so that the vapors can escape upwards from each, while the limber holes, provided for drainage, allow the fluid to gravitate towards the keel. The upper holes would generally be required in the longitudinal frames only, and they should not be large enough to allow any considerable ascending of the liquid when the ship is rolling. The petroleum vapors are heavy and tend to accumulate near the surface of the liquid, and, if the tank is empty, near the bottom of the tank. For the removal of this vapor, which may become dangerous under unfavorable conditions, pump suctions should be led to near both the top and bottom of the tank. Space must be allowed above the oil for its expansion when warmed by the ship or sea water outside it.

The open tops of the ventilating pipes should be protected from sparks or lighted matter falling into them.

SPECIFICATIONS FOR OIL-STORAGE COMPARTMENTS.

Fuel Oil.—The rivets shall be spaced not more than three and one-half diameters apart in the same row, and no laps, seams or butts shall have less than double riveting. All stiffeners, angle irons, etc., on the outside of tanks shall be

caulked, and laps, seams and butts shall be caulked on both sides where directed. All oiltight bounding angles shall have bosom pieces to take at least four rivets on each side of butt.

Special consideration shall be given to securing the most superior riveting, fitting, etc. Three-ply reveting shall be avoided whenever practicable. All holes shall be punched small and properly reamed, or, if directed, shall be drilled in place.

The above requirements apply to all fuel-oil tanks. In addition, those compartments specified to be fitted as storage tanks shall be provided with the necessary trunks, cofferdams, sheathing, sounding tubes, means for indicating the quantity of oil, air escapes, and all necessary valves, piping, etc., for pumping and storing oil fuel.

The compartments shall be fitted with warning signals to give distant, and, if directed, local warning when the oil in filling enters the trunk.

All outlets and inlets from the tank shall be through the top, without horizontal sections in the tank. Air-escape pipes shall be fitted to the tanks, cofferdams and expansion trunks; these in all cases to exceed the size of the filling pipes and to extend above the main deck. These pipes should have goose-neck heads and be fitted with wire mesh and closing valves.

All pipes, except ventilation, overflow and oil-escape pipes, shall extend to the bottom of the tanks, to prevent the accumulation of gas.

There shall be provided connections for steaming out the tanks, and salt-water connections for cleaning. Overflow pipe shall be fitted for use in cleaning out.

All valves shall be operated from the deck next above the speed-trial water line, or as directed.

To subdivide the area of fuel-oil surface light, non-watertight covers shall be provided for the access and lightening holes in such non-watertight longitudinal as may extend through the tanks. Special care shall be taken to thoroughly protect and insulate any electric wires in the vicinity of the installation and to prevent any sparking.

Surfaces exposed to the oil are to be covered with shellac or linseed oil, as directed, or other approved covering.

Lacquered leather or other approved material shall be used in the manhole covers and elsewhere, as directed.

In Admiralty practice with oil tanks for destroyers, great care is taken in construction, as, on account of the lightness of plating, no caulking can be done. The tanks are tested for watertightness and are then filled with oil under a 24-foot head, and must show no leaks in a 72-hour test.

With oil piping great care is necessary to obtain good joints. In the best practice they are all machined. British practice seems to be to use copper piping and flanged joints.

Settling or Separation Tanks.—The presence of water, which is very common in small or large quantities in the oil fuel, increases the difficulty of obtaining efficient combustion and considerably reduces the evaporative power. Every 1 per cent. of water contained in the fuel reduces the evaporative power 2 per cent. The presence of water is generally indicated by the color of the flame. With satisfactory combustion the flame is an opaque, dazzling white for about six inches from the end of the jet, then gradually becomes semi-transparent, then at mid-length becomes violet, and towards the end changes to red. When mixed with a moderate amount of water no violet is observed, and the red becomes darkened and fringed with smoke. If only a small proportion of water is present, instead of the dark red, the end of the flame is white, and this is probably due to acetylene.

The densities of the oil fuel (.9) and water (1) do not differ very much. The lighter oil does not readily rise to the surface, and it is necessary to accelerate the separation by heating or other means. To do this the fuel is pumped into settling tanks, each containing about 12 hours' supply. A coil of piping heated by exhaust steam may be made to float a few inches below the surface of the oil, and the heat set up a circulation of the oil and water past it. As the oil and water have different expansions they separate more rapidly when heated. The separation by heating and circulation leaves the lighter

oil at the surface while the heavier water gravitates to the bottom of the tank. A gage glass shows the relative levels of the oil and the water; the water is drawn off from the bottom of the tank.

When separating oil from water by heating it only sufficient temperature for the purpose should be allowed in the heating pipe of coil. Vapor is given off by the fuel at a temperature a little below the flash point, and any great amount of heating is dangerous.

STORAGE ON SHORE.

10,000-Barrel Steel Storage Tank for Refined Oil.—Texas Oil Company.

Dimensions: 50 feet diameter; 30 feet high.

Built of soft open-hearth tank steel.

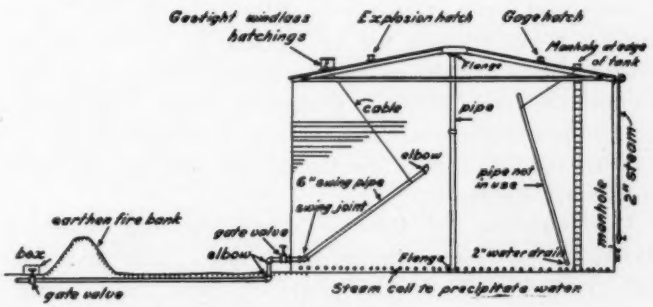
Bottom plates, $\frac{1}{4}$ inch in thickness.

First ring of shell $\frac{5}{16}$ inch in thickness.

First ring of shell joined to bottom of tank with 3-inch by 3-inch by $\frac{3}{8}$-inch angle bars.

Second ring of shell $\frac{5}{16}$ inch in thickness.

Third, fourth, fifth and sixth rings, No. 6 U. S. gage.



TYPICAL OIL-STORAGE TANK OF THE TEXAS CO.

Roof built of No. 10 U. S. gage steel sheets resting on an angle bar at the top of the sixth ring. The rafters are supported at the center upon a circular rest plate, and an iron-

pipe column, 8-inch diameter, and at the sides with ¼-inch flanged gusset plates riveted to the sixth ring. The entire tank and roof caulked to be oil and gas tight.

These tanks are generally fitted with one manhole through roof of tank, and ladder leading down therefrom. One manhole in the first ring, one or more gaging hatches. One 6-inch swing pipe for loading or discharging. This swing pipe may be raised or lowered by means of windlass and cable, all of which is operated from gas-tight hatch in roof of tank. The object of swing pipe is to enable it to discharge from any depth beneath the top of the tank. It is desirable to discharge tank with swing pipe just below the surface of the oil, since water and moisture carried in suspension in the oil will settle at the bottom. Oil pumped from near the surface will be free from same. There is also a 2-inch swing pipe used for drawing off this water from the bottom of the tank. In crude-oil tanks a steam coil is sometimes used to precipitate more rapidly this water or moisture.

Tanks are provided with "free steam" pipes leading to the top of the tank. In case of fire in the vicinity, steam is allowed to enter and fill the tank above the surface of the oil, thereby smothering any tendency of the gases to ignite. These steam pipes are generally 2-inch pipe having "goose neck" for expansion at the top, as shown. Tanks are provided with one or more explosion hatches, located on the roof. Earthen fire banks are built around the tanks; their relative location and height is so designed to form a reservoir of cubic contents equal to that of the tank. This as a fire protection to adjacent equipment.

The Standard Oil Company's extinguishing system at Bayonne, New Jersey, consists of two tanks from which pipes lead to small mixer chamber and from there lead to surface of the oil that is burning.

The contents of the tanks are as follows:

1st tank: Glue, glucose, bicarbonate of soda, salicylic acid and water.

2d tank: Aluminum sulphate, sulphuric acid and water.

The idea being that the giving of large quantities of CO_2 would smother the flames, which are kept in contact with the surface of the burning oil by the suds formed with the glucose.

This combination effectually extinguishes a fire, but it requires large quantities of it, which practically excludes it from use on board ship.

Pyrene Extinguisher.—This has for its base carbon tetra chloride (CCl_4), which is a heavy liquid that evolves large quantities of non-combustion supporting gas when heated to about 200 degrees F., the liquid simply being sprayed on the flames.

ADVANTAGES OF LIQUID FUEL FOR MARINE USE.

1. *Reduction of Weight and Space.*—Fuel oil has a greater evaporative efficiency than coal in the ratio of about 14 to 9. Also while a ton of coal occupies about 43 cubic feet, a ton of oil occupies about 39 cubic feet. For the same amount of fuel the bunker space could be reduced to about five-eighths that required for coal, or, using the same space now given for coal, the steaming radius could be increased from fifty to one hundred per cent.

2. *Reduction in the Fireroom Force.*—A battleship now having a complement of about 220 firemen and coal passers could have it reduced to 50 firemen and coal passers, which would be more than enough for operating and cleaning purposes. The saving in pay and subsistence would be a considerable item, and the space now taken up for washrooms, lockers, etc., would be available for the remainder of the crew, greatly adding to their comfort, or could be utilized for stores. The saving of fresh water would be considerable, and, from a point of discipline, a reduction in the fireroom force of almost any ship would be an attractive feature.

3. *Elimination of Ashes and Dirty Fires.*—No coal and ash-handling gear with the resulting dirt.

4. *The Rapidity and Cheapness with which it can be Brought on Board.*—An example of this is given on the *Delaware,*

where oil has been taken on board through a single hose at the rate of 100,000 gallons (about 338 tons) per hour. Where an entire ship's company is required to coal ship and at least one day is taken to do it, with an extra day to clean up, oil could be put on board our largest ship in six hours at least, and not more than ten or twelve men would be required to do the work, and no dirt would result. It would also afford an easy means of getting fuel on board at sea while ships are underway.

5. *The More Perfect Combustion of Liquid Fuel Prevents the Formation of Residuals and Smoke.*—A steady pressure can be kept with much greater ease, as furnace doors do not have to be continually opened, with consequent inrush of cold air. There is much easier control of fires, permitting variation in power. The boiler tubes being free from soot and practically always clean are in the best condition for transmitting heat. There is also but little waste of fuel on starting up or shutting down.

6. *Portions of a Ship which are now Unsuitable for Coal-Bunker Purposes through Inaccessibility are Available for the Storage of Oil Fuel.*—The double bottoms can be used and the trim of the ship can be easily preserved. There need be no such thing as a list to port or starboard through being obliged to use coal out of one bunker more than another. A steam pump will readily remove oil from port to starboard or vice-versa, when the same expedient is impossible with coal.

7. *We are in a Better Condition to Use Oil Fuel than any other Nation,* since the greater part of the world's production of petroleum comes from the United States.

COMBUSTION.

For complete combustion of oil fuel the following conditions are necessary: (1) the gases must be kept at the ignition temperature long enough to insure their burning; (2) there must be a sufficient supply of air; (3) the air must be intimately mixed with the fuel. The first condition

is fulfilled by having proper furnace arrangements—ample combustion space, and suitable refractory linings. During the early stages of combustion of any fuel the gases of a highly volatile nature distil at a low temperature, rise rapidly, hug the boiler, enter the tubes or flues and pass out unconsumed. The combustion chamber should therefore be arranged with fire brick so that the flames may be retarded, diffused and distributed, and the velocity impeded. In burning liquid fuel the oil is practically converted into gas, vapor and floating particles of finely-divided carbon before combustion begins, or, at any rate, before any large amount has been consumed. The first effect of the heat is to dissociate the carbon from the hydrogen. The carbon being left alone is carried along in a current of gases, and, unless it has sufficient time to take up its amount of oxygen for combustion, it is going to be left as soot, or carried along with the excess of oxygen with which it should have been united before being cooled below the temperature of ignition, and forms smoke. Soot once formed is almost impossible to burn, and the aim is to prevent its forming rather than to try to burn it once it has formed. Suitable refractory linings in the furnace also help to keep the temperature high, and the length of flame from a burning hydrocarbon is largely determined by the intensity of combustion. A well-mixed gas burning at a high temperature will produce a short flame, whereas the same gas burned in water-cooled flues at a much lower temperature will produce an exceedingly long flame. The impact of the flame on a cold surface, such as a boiler plate or a water tube, with a temperature of about 400 degrees F., checks combustion and also produces smoke, as there is soot deposited. Experiments show that ample combustion space and suitable linings, by giving time for diffusion and keeping the temperature high, diminish the volume of the flames.

In an oil-burning plant, since the admission of air is under complete control and the fuel is burned in fine particles in close contact with the oxygen of the air, only a small excess of air above that actually necessary for the complete combus-

tion of fuel is required. Calculations as to boiler performance are simplified with oil fuel, as practically complete combustion is secured in a well-designed furnace, the carbon and carbon monoxide usually being burned to CO_2. It is possible to obtain a fair notion of the excess air from a mere determination of the amount of CO_2; that is, assuming all hydrogen having been burned to H_2O and all carbon to CO_2. Any simple formula involving the element CO_2 must be dependent on an assumed percentage of hydrogen in the oil fuel; but inasmuch as the hydrogen contained is fairly uniform for any given grade of oil, there is but little error in such an assumption.

POUNDS OF AIR PER POUND OF OIL, AND RATIO OF AIR SUPPLIED TO THAT CHEMICALLY REQUIRED.

Per cent. of CO_2 by volume, as shown by analysis of dry chimney gases.	C, 84; H, 13; S, 0.8; N, 0.2; O, 1; H_2O, 1 per ct.		C, 85; H, 12; S, 0.8; N, 0.2; O, 1; H_2O, 1 per ct.		C, 86; H, 11; S, 0.8; N, 0.8; O, 1; H_2O, 1 per ct.	
	Pounds of air per pound of oil.	Ratio of air supply to chemical requirements.	Pounds of air per pound of oil.	Ratio of air supply to chemical requirements.	Pounds of air per pound of oil.	Ratio of air supply to chemical requirements.
4	51.40	3.607	51.93	3.704	52.45	3.803
5	41.31	2.899	41.71	2.975	42.12	3.054
6	34.58	2.427	34.90	2.490	35.23	2.554
7	29.77	2.089	30.04	2.143	30.31	2.198
8	26.17	1.836	26.39	1.883	26.62	1.930
9	23.37	1.640	23.56	1.680	23.75	1.722
10	21.12	1.482	21.29	1.518	21.45	1.555
11	19.83	1.391	19.43	1.386	19.58	1.419
12	17.76	1.246	17.88	1.276	18.01	1.306
13	16.46	1.155	16.57	1.182	16.69	1.210
14	15.36	1.078	15.45	1.102	15.55	1.127
15	14.39	1.010	14.48	1.033	14.57	1.056

Under the present systems of firing the amount of CO_2 present in the flue gases is often as low as 4 to 5 per cent. The presence of CO_2 under an ideal and uniform load can be maintained as high as 13 per cent.

The various text books give from 15 to 18 pounds of air per pound of oil as the minimum necessary to supply the necessary oxygen for combustion with from 5 to 10 per cent. excess in best practice, while some claim that from 50 to 100 per cent. is necessary, depending on the system used and the

condition of the furnace in which it is burned. An excess of air causes a loss not only on account of the direct loss in heating the added air to the temperature of the flue gases, but there is a secondary loss due to the fact that, corresponding to an excess of air, there results a higher flue temperature not only for the actual amount of air necessary for combustion, but for all such excess air. Assuming 10 per cent. excess as the ideal amount of excess air, complete combustion and boiler radiation loss of 3 per cent., the following boiler efficiency and saving in fuel was calculated by Mr. C. R. Weymouth and published in the Journal of American Society of Mechanical Engineers, 1909.

Excess air supply, per cent.	10	50	75	100	150	200
Assumed temperature of escaping gases, degrees F.	400	450	475	490	Over 500	Over 500
Corresponding ideal efficiency of boiler, per cent.	84.2	80.27	77.66	75.22	Under 70.94	Under 67.09
Possible saving in fuel due to reduction of air supply to 10 per cent. excess, expressed as per cent. of oil actually burned under assumed conditions.	0.0	4.67	7.78	10.68	Over 15.76	Over 20.32

Method of Introducing Air.—On the trial trips of the destroyers it was found that in order to burn oil properly there must not only be a constant relative supply of oil and air, but that the air must enter the furnace with the oil at a relative velocity. Experiments were made by increasing the velocity of the air without increasing its volume, which proved that for any furnace the proper volume of air can have its velocity changed so that all vibration ceases and quiet combustion takes place. Increasing the velocity of air also decreases the amount of oil that might otherwise be used.

Method of Introducing Oil.—To intimately mix the oil and fuel the fuel must be broken into fine particles or atoms and each atom surrounded with a sufficient amount of air. The object of breaking up the fuel is to increase the surface area by minutely dividing it into a spray of the greatest number of particles of the smallest size, and then to allow

the air required for combustion to become intimately mixed with the fuel. The finer the division is, approaching very closely to the consistency of a fog when using a good atomizer, the greater the surface area becomes, and the more intimate is the mixture with the air supplied for combustion. This may be accomplished by gasifying or atomizing. Gasifying where the fuel is heated and converted into vapor before going to the furnace has not been found satisfactory, and atomizing is recognized as the best method of burning.

Systems of Atomizing Oil.—The necessary impelling force for atomizing is generally supplied by—

(1) Delivering the oil under pressure, as in the mechanical-spray type. This is the type used in the Navy, and the oil is delivered in conical form, a whirling motion being imparted to it. The air comes in around the burner in an enveloping cone. In some cases the cone is given a whirling motion opposite to that given the oil. The merit of this feature is in dispute. In mechanical atomization systems it is necessary to put the firerooms under air pressure in order to force the air into the furnace along with the oil and have it present in a mechanical mixture such as will burn readily. This makes it necessary to run under forced draft under all conditions, which is decidedly objectionable in port. Pressures used in mechanical burners vary from 100 to 200 pounds.

(2) Inducing a flow by a jet of steam which is frequently superheated.—This system is almost generally used on shore, but very little for marine purposes. The objection to its use is the amount of water used for atomization, most authorities giving from 2 to 3 per cent. of the total amount evaporated, while the Navy Fuel Board estimated it as high as $4\frac{1}{2}$ per cent. The burners are designated as outside or inside burners, according as the steam surrounds or is surrounded by oil before leaving the burner. It has the advantage of heating the oil, but water contained frequently extinguishes the burner.

(3) Inducing a flow by a jet of compressed air which is frequently heated.—This system is employed to a great extent in the merchant marine, and is very efficient. When the

air is heated a shorter flame is usually obtained than when using steam or mechanical means, and nearly any kind of oil can be used. It has the disadvantage of cost of compressors, their upkeep, noise and extra weight.

Heating the Oil.—In all systems the oil that is to be immediately burned is first heated. Heat deprives the oil of its viscidity, and has the further advantage that any water in the oil more readily separates out of hot oil than cold oil, first, because the heated oil, being more limpid, offers less resistance to the freeing of water, and, second, there is a greater expansion of oil than of water due to the heat, and the water gains a relatively greater specific gravity. The oil should not be heated above the temperature best suited for most efficient combustion. See paper by E. H. Peabody, in JOURNAL A. S. N. E., May, 1911.

Complete Combustion.—The guides to complete combustion are: (1) an analysis of the smoke-pipe gases; (2) the amount of smoke given out by the stack: (3) the appearance of the flame itself.

By analyzing the smoke-pipe gases and obtaining the percentage of CO_2, a positive guide is at once obtained as regards combustion. The percentage of CO_2 with careful adjustments can be kept as high as 13 per cent.

The amount of smoke given out is a good guide, but not a positive one, as absence of smoke does not indicate complete combustion. To readily observe the amount of smoke from the fireroom an electric light may be placed in the boiler in the back portion just below the uptakes. A round deadlight is opposite in the front end of the boiler, and reflecting mirrors enable the attendants in the fireroom to see into the space back of the deadlight. Any smoke hides the lamp or dims it.

Probably the best method is to observe the flame itself. Peep holes are provided in the furnace so that the interior can be seen. With good combustion the gases become clear, clean, brightly incandescent, red, and the opposite end of the furnace shows up clearly. So long as there is smoke formation the opposite brickwork cannot be seen. As soon as

combustion is perfect it appears clear and bright red, and the air should then be gradually cut down in quantity until an occasional streak of dark colored gas begins to show, thus proving that under the conditions of the furnace the air has then been reduced to a possible minimum.

If looked into through a piece of violet-colored glass the interior of the combustion chamber with its brick linings should show a light lavender color, indicative of perfect combustion, with the production of actinic rays, indicative of high chemical action. A chilled fire, such as is produced where a boiler is placed close upon the furnace of a coal fire, will show very little light through a violet glass, its flames being cut down from several feet in length to a few inches only in many instances, the flames of yellow and reddish intensity being resolved into streams of dim colored gas which throw off no light of sufficient actinic power to penetrate the glass.

Oil as Auxiliary to Coal.—The use of solid and liquid fuel in the same furnace together modifies the condition for each fuel.

For coal the efficiency of combustion should be better, and for oil the heat should be better utilized.

Combustion on the grate may be imperfect, but the oil atomizer so mixes up the gases from the grate with the air admitted through and above it that combustion should be much improved and the excess of air would be used by the oil.

This would be true where furnaces are arranged so that a sufficient amount of air could be obtained to burn each fuel, but with furnaces arranged to give the best results with coal alone, using oil as an auxiliary only in special cases, the condition will not allow of sufficient air supply to properly consume the oil and develop its full advantages, while a dense flame of oil burning above the coal will keep the air away from it and deaden the surface combustion.

Experiments on the *Delaware* proved that it is difficult to burn the oil because sufficient air did not get to it, the amount

being limited to the amount of air that gets in alongside the burner and through the furnace doors, which must be left open to accomplish anything. It resulted generally in a great deal of smoke and uneconomical performance.

INSTALLATION FOR U. S. BATTLESHIP "FLORIDA."

All oil is carried in ten double bottoms under the engine rooms, five on each side. A suction pipe, 4 inches inside diameter, leads from near the bottom of each compartment to a manifold in the engine room on its own side of the ship. Each bottom has a ventilating pipe leading from the highest part of the oil compartment up the engine-hatch casing above the main deck. It is fitted with a quarter turn on outside of casing, and arranged with a vertical pipe to extend down to about 3 inches above the deck. Wire-mesh gauze is fitted between the flanges of the air escape some distance above the oiltight compartment, also at the flange connecting to hatch casing above the deck. A coarse wire-mesh gauze is fitted to the deck opening.

Sounding tubes, $1\frac{1}{2}$ inches in diameter, are fitted in each compartment and arranged with valves, caps and locking devices.

The supply pipe is 6 inches in diameter and is led under the main deck from side to side of the ship, terminating in a connection at each side of the ship, for a 6-inch hole. A 6-inch branch leads from the athwartship pipe to the connecting pipe between manifolds mentioned above. A cut-out valve is fitted in the filling pipe adjacent to the pipe connecting the manifold. An overflow pipe is provided to the filling pipe just above the cut-out valve.

Steam connections lead to the top of each double bottom to space above oil, also connections, $2\frac{1}{2}$ inches in diameter, are taken from sea valve for pump discharge for cleaning oil compartments.

Fuel-oil compartments are filled to only 95 per cent. of their

net capacities and warning devices are installed to indicate when filled to that extent.

Oil is handled by two pumps, 10-inch by 7-inch by 12-inch, one near each manifold. They discharge through a strainer in engine room to a 3½-inch pipe leading to firerooms. This pipe branches to two 2½-inch pipes at after fireroom bulkhead, each pipe leading forward outboard of the boilers. An oil heater is provided in firerooms, one for every two boilers, and provision is made to by-pass the oil around the heaters as desired.

INSTALLATION ON TORPEDO-BOAT DESTROYERS "PAULDING" AND "DRAYTON."

The system consists of storage tanks, tank-supply pumps, settling tanks, oil-service pumps, heaters and burners.

From the storage tanks the oil fuel is pumped by two Blake simplex oil-supply pumps to the settling tanks, of which there are four, two alongside of each boiler compartment. The oil-supply pumps are connected so that they can discharage from one storage tank into another, so that they can discharge oil to another vessel, or pump oil on board from a barge or oil carrier.

From the settling tanks the oil is pumped by means of the oil-service pumps, two in each boiler room. These pumps discharge through oil heaters, strainers, and then to burners. A small hand pump is supplied in each boiler compartment for starting up when there is no steam on vessel.

Storage Tanks.—These tanks are fitted with 4½-inch vents leading to non-return mushroom openings on deck. There is a deck-filling connection, and provision for blowing out tanks by means of steam hose is also provided.

Settling Tanks.—These tanks have 4½-inch vents leading to deck, are each fitted with a gage glass, and have a small hand pump for pumping overboard water collecting in bottom of tank. Steam-heating coils are fitted in the tank near suctions.

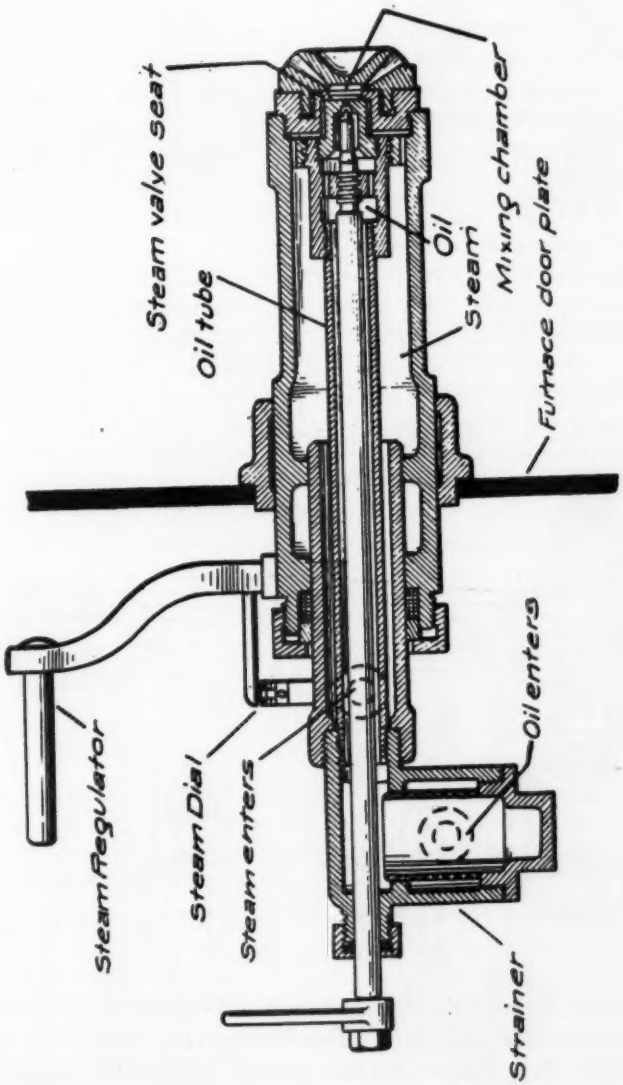

Steam valve seat
Oil tube
Oil
Steam
Mixing chamber
Furnace door plate
Steam Regulator
Steam Dial
Steam enters
Oil enters
Strainer

Fig. 1.

Oil Heaters.—There are two heaters in each boiler compartment. The heating surface of each heater is 131 square feet, there being 149 $\frac{5}{8}$-inch steel tubes, 18 B.W.G. thick. The heaters are built in a manner similar to the feed-water heaters, but oil passes three times through the tubes. Live steam is used as the heating agent. The heaters are drained to traps and the discharges are arranged so that they can be led to a testing tank to observe whether there is any oil leakage to steam side of heater.

Figure 1 shows the Lassöe burner, which uses steam as the atomizing agent. This burner has been installed in a large number of merchant steamers and great efficiency is claimed for it. Proportion of steam used for burners to that generated is claimed to be as low as $\frac{1}{2}$ to $\frac{3}{4}$ of 1 per cent., as determined by careful tests covering a period of two months.

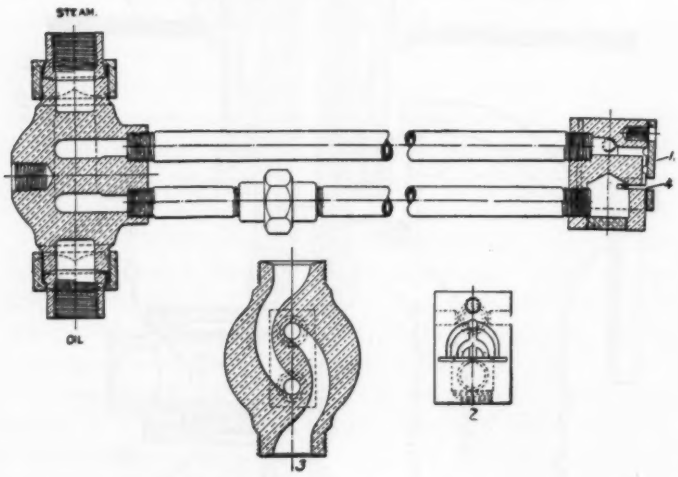


Fig. 2.—INGRAM BURNER.

Figure 2 shows the Ingram burner, which uses either steam or air as atomizing agent. This burner was installed on the U. S. S. *Delaware* and has been used on some of the torpedo-boat destroyers in port.

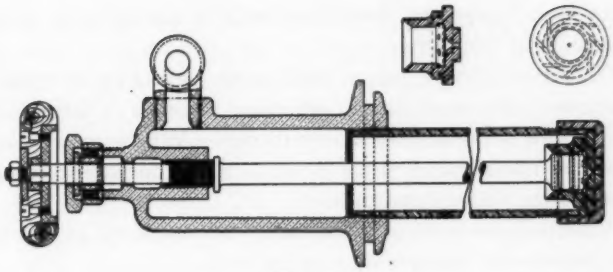


Fig. 3.

Figures 3 and 4 show the Normand burner and tuyere installed on torpedo-boat destroyers *Paulding, Drayton* and *Trippe,* built at the Bath Iron Works. No serious trouble has been experienced in use of these burners, oil as heavy as 25 degrees Baumé has been used successfully. Firerooms are put under pressure and air is admitted through tuyeres. Tuyeres have no adjustment for changing velocity of air.

Figure 5 shows the Thornycroft burner installed on torpedo

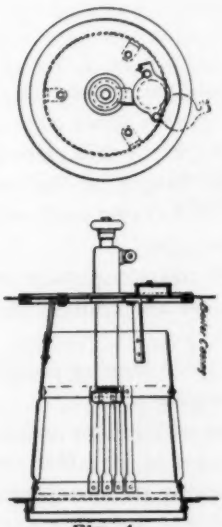


Fig. 4.

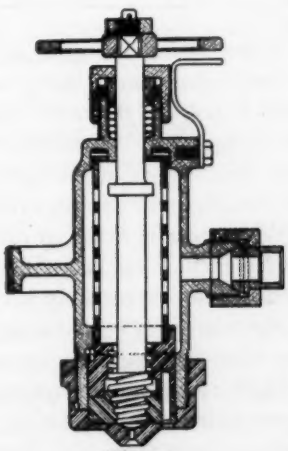


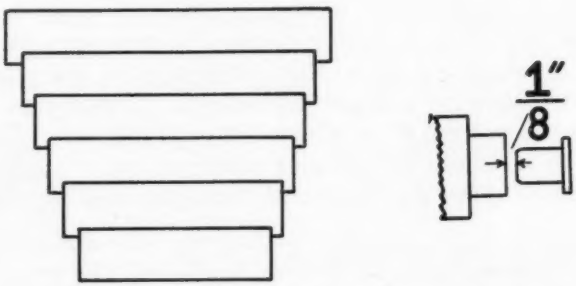
Fig. 5.

boats *Roe, Terry* and *Monaghan,* built at the Newport News Shipbuilding Company.

The whole of this gear is made with great care as regards joints, etc., and steel is used wherever possible. Distinguishing feature is the tapered screw fittings. Where castings are necessary the best gun metal is used. Air is admitted through cones of the shape shown below.

A certain amount of adjustment is necessary to fix the best position of burners as regards cones.

Experience of shipbuilders in their trials gave the best position as $\frac{1}{8}$ inch from edge of cone.



When fitting the gear all joints should be scraped metal to metal, the flanges being racked off from the inner edge of the bolt holes to the outer edge of the flange. A thin piece of oiled paper may be inserted between the flanges on the suction side. Pipes in the system are of solid-drawn steel with the flanges screwed and expanded on the pipe.

When gear is fitted an oil test of 400 and 150 pounds per square inch is applied for three hours on the delivery and suction sides respectively.

Starting Fires.—The contractors use the following procedure when lighting fires for steaming purposes.

Place a small amount of waste saturated with fuel or paraffin in one of the cones, preferably near the center of the boiler, and ignite to heat the cone. The pipe connecting the distributing box to the burner should also be wrapped with a small amount

of waste saturated with oil and lighted to heat the oil to the necessary temperature before being sprayed.

Start the hand pump and maintain a steady pressure of from 250 to 300 pounds. As soon as sufficient steam is raised pass it through the oil heater and start the fans and oil pump gently. When sufficient steam is generated to be certain that the fuel-oil pump will keep working the hand pump may be dispensed with. The temperature of the oil should be carefully watched and maintained at about 200 degrees F. After steam is raised, should black smoke be observed at the funnel, more air is necessary, and the speed of the fans should be increased; should the fans be running too hard white smoke will be seen coming from the funnel.

When the oil is being efficiently burnt very light smoke should be observed at the funnel during each stroke of the pump when the pressure is highest, and this should disappear when the pressure drops at the end of the stroke.

If small, fiery flashes are observed in the furnace this means that particles of unevaporated oil are being thrown into the furnace, and will probably be due to the flame not curling back into the cone. This can be remedied by closing the sprayer down slightly until the cone gets hotter and the flame flares out evenly from the burner tip.

Fig. 6.

Figure 6 shows a burner invented by Mr. E. H. Peabody of the Babcock & Wilcox Company, New York. Extensive experiments were conducted with this burner on a Babcock & Wilcox boiler. The air device finally adopted is shown in figure 8 (a). Very heavy oil, 15 degrees to 20 degrees Baumé, was burned without difficulty. This air device was adopted after trials with air cones which were not altogether a success.

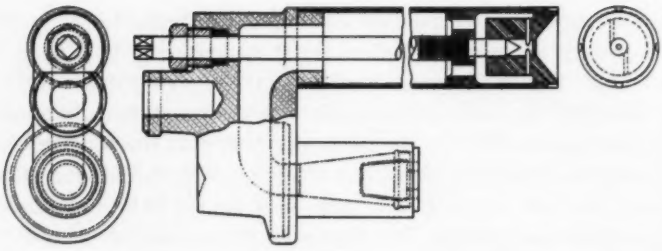


Fig. 7.

Pressure used with 200 to 250 pounds and burner was efficient down to pressure of about 50 pounds.

Figure 7 shows the Fore River Shipbuilding Company's burner. It is installed on the torpedo-boat destroyers *Perkins, Sterett* and *Walke*. The tuyere used is very much like

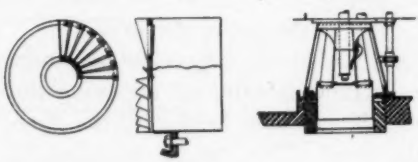


Fig. 8.

that used with the Normand burner shown in figure 4. On the U. S. torpedo-boat destroyer *Sterett* excessive vibration in the boilers took place above a speed of twenty-five knots. There is also considerable variation of pressure in the oil lines. These difficulties have been met by changing the air

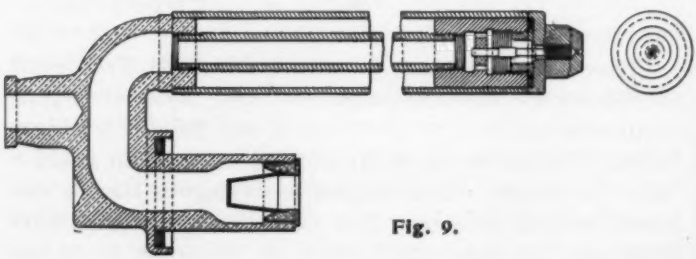


Fig. 9.

cones to get better regulation of the air and by installing air cushions on the pipe lines.

Figure 9 shows the Koerting burner installed on torpedo-boat destroyers built at the Cramps and at the New York Shipbuilding companies, and is supplied by them to the following U. S. torpedo-boat destroyers: *Burrows, Mayrant, Warrington, Ammen, McCall* and *Patterson.* The air-regulating

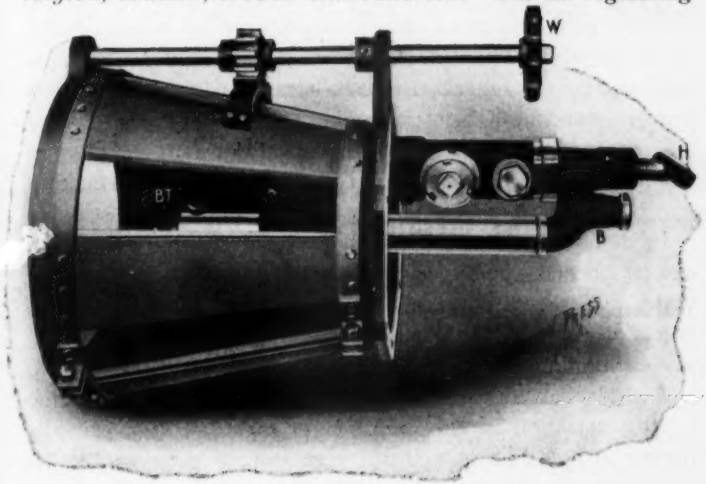


Fig. 10.

device is shown in figures 8 and 10. Opening in slots can be varied at will, thus changing the velocity of air, which is claimed by inventors to be of primary importance. On builder's trials of destroyers built at Cramps the Peabody air device was installed together with the Koerting device, and better results were obtained.

OPERATION.

Shore Plant, Pacific Electric Co., Los Angeles, Cal.

Steam atomizing is used in this plant, and a plan was developed for firing eighteen boilers, averaging three burners per boiler, totaling fifty-four burners, with central hand control of oil pressure.

The operator was stationed near the oil pumps, which were run at a practically constant speed. In front of the operator were the oil-pressure gages connecting the oil main and the steam-pressure gage connecting to steam main. The operator's duty was to maintain a uniform steam pressure on the boilers by opening or closing a bleeder valve on the oil-pump discharge line, thus increasing or decreasing the pressure in the oil main and simultaneously the rate of firing of all boilers. The operating crew of the boiler room for each shift consisted of the one operator controlling the oil pressure and one water tender.

At the Redondo plant of the Pacific Light and Power Company, near Los Angeles, California, an automatic system was used.

In this system all individual-burner valves, both steam and oil, are opened wide or nearly so, and all burners are operated under full pressure in the respective mains. In the larger plants all dampers are connected to a common rock shaft and move simultaneously.

A slight variation in the steam pressure on the boilers, due to any variation in the demand for steam, is the primary means of control for a steam regulator or governor which varies the oil pressure at the oil pumps and in the oil main. Corresponding to an increased pressure in the oil main there is an increase in the amount of fuel fired and a rise in boiler steam pressure, and corresponding to a decrease in pressure in the oil main there is a decrease in the rate of oil fired and a lowering of the boiler steam pressure; this regulator thus maintains a uniform steam pressure on boiler at all loads within the governing limits. The variation in pressure in the oil main is the secondary means for controlling the supply of steam for atomizing purposes and also for controlling the supply of air for combustion.

The supply of steam to burners is controlled by regulating the pressure in a separate low-pressure main common to all burners, the pressure in this main having a certain predetermined relationship to the pressure in the oil main and being

controlled by a ratio regulator. By means of a specially-constructed diaphragm regulator the opening of the boiler dampers is made to increase or decrease with a corresponding variation of pressure in the oil main, the change in damper opening, in turn, governing the supply of air for combustion.

For Torpedo-Boat Destroyers (New York Ship Building Company).

Lighting Fires.—(*a*) Use a burner with smallest tip furnished, which is 1½ millimeters; place in center of boiler on upper row, start hand pump, and when pressure on oil gage shows 100 pounds open burners and light with torch provided. Some little difficulty in igniting may be found when not using the charcoal heater, but it can be done with cold oil.

(*b*) As soon as sufficient steam is raised in the boiler start a blower and bring the air pressure up to an equivalent of 3 inches of water pressure, then turn steam on the oil-service pump.

(*c*) One burner of 1½-millimeter opening being in use, the other burners, which are 2.3-millimeter, can be cut in, one at a time, always increasing the speed of the blower so as to maintain a proper air pressure, which should be about 4 inches when all burners are in use and 100 pounds oil pressure is shown on oil gage at the boiler front. Under these conditions air registers should be closed to about ⅝-inch mean opening. Oil heaters should be cut in and a temperature of 200 degrees should be used at burners.

(*d*) If more steam is required the oil pressure should be gradually increased from 100 pounds to 150 pounds on all burners. This will require the blower being speeded up in advance of the oil pressure so that when all burners are in use and the oil pressure is about 150 pounds an air pressure of about 5 inches will suffice. Air register should be open about ⅝ inch under these conditions; likewise two blowers will be needed, on account of the loss of air which passes up through the other boiler which is not in use.

(*e*) If vibration occurs when a burner is being cut in stop it by immediately cutting out the burner, or two burners can be cut out, until the air pressure can be raised, after which it will cease.

(*f*) Under no condition should vibration be allowed to continue, as it is due to the air not having sufficient velocity.

An increase in air pressure above 5 inches will be found desirable when working under an oil pressure above 130 pounds, and under no circumstances should the air pressure be restricted, as it can do no harm and usually proves beneficial. In fact, a high-air pressure will enable less air to be used per pound of oil.

(*g*) When necessary to light up the other boiler in the same compartment it will be necessary to first increase the speed of the blower, then open up air register to about $\frac{3}{8}$ inch, next turn on oil at burners, one at a time. In this manner all burners may be cut in and no vibration will be noticed. Don't open up air register until ready to cut in its own oil burner. Keep watch on the water gage, which records the air pressure, until satisfactory conditions are obtained, after which no trouble will be experienced without change of condition. As heated oil is being supplied to this boiler no difficulty will be found in lighting.

(*h*) With one boiler only in use the other boiler should be closed up as tightly as possible, so as not to waste the air in the fireroom.

(*i*) If two boilers are in use in one compartment they should always have the same number of burners in use, and the air register should be kept in similar positions.

(*j*) When less than the full number of burners are in use in any boiler the upper wing burners are the most important to cut out; then the lower wing burners are next in importance to leave out; this keeps the oil spray away from the cold tubes and prevents carbon being formed on tubes.

(*k*) At full power, when it becomes necessary to turn ventilators away from the wind, this should be done only after the blowers have been speeded up enough to increase the

air pressure about $\frac{3}{4}$ inch. About thirty revolutions should accomplish this. After the ventilators have been trimmed it will be found that the pressure has dropped back to about what it was before the blowers were speeded up.

Emergency.—When emergency occurs, requiring the cutting out of a boiler, close master valve, shutting off all burners. If two boilers are in use and only one is to be cut out the fireman on duty should close each oil burner, at the same time shutting off the air registers. This air pressure should be kept up until all burners in the one boiler are cut out, then the revolutions of the blowers reduced so as to bring down the air pressure to the required amount.

It will be found that burners work very well with air registers open about $\frac{5}{8}$ inch and an air pressure of from 4 to 6 inches, depending on the oil pressure carried, viz: 4-inch air pressure will do for 100 pounds oil pressure, while 6-inch air pressure will be found desirable when the oil pressure is 175 pounds.

U. S. TORPEDO-BOAT DESTROYER *MONAGHAN.*

By U. T. Holmes, Commander, U. S. Navy.

CONTRACT TRIALS.

The *John R. Monaghan* is the fourth to be tried of the five torpedo-boat destroyers authorized by Act of Congress, approved March 3, 1909. The contract for her construction was made with the Newport News Shipbuilding and Dry Dock Company on June 23, 1909. The keel was laid June 1, 1910, and the vessel was launched February 18, 1911.

In building the *Monaghan* the builders followed closely the plans of the *Roe* and *Terry*, so that the description of these vessels which appeared in the February, 1911, number of the Journal applies also to the *Monaghan.*

SCHEDULE OF TRIALS.

The *Monaghan* was subjected to the following trials:

(1) Standardization trials over a measured course, at speeds of, approximately, 12, 16, 20, 23, 24½, 27½ and 29½ knots, and at the highest speed attainable. Three runs were made at each speed, except the maximum. Five runs were made at the highest speed.

(2) A four-hour full-speed trial.
(3) A four-hour 27½-knot trial.
(4) A four-hour 25-knot trial.
(5) A four-hour 20-knot trial.
(6) A four-hour 16-knot trial.
(7) A four-hour 12-knot trial.

The standardization trials were begun on the measured course at Delaware Breakwater, May 23, 1911. Owing to thick weather these trials were interrupted, but they were

completed on the morning of May 24, 1911. The sixteenth run was thrown out on account of the ranges becoming obscured. The runs at 27½ knots were repeated on May 24th, the runs at that speed on May 23d having been made at too light a displacement.

The results of the standardization trials are given in the following table of standardization data and are also shown on the appended curve.

STANDARDIZATION DATA.

Run No.	Starb'd shaft.		Center shaft.		Port shaft.		Average.	Total.	Speed.
	R.P.M.	S.H.P.	R.P.M.	S.H.P.	R.P.M.	S.H.P.	R.P.M.	S.H.P.	
1	306.25	350	279.84	163	307.44	229	290.78	742	12.304
2	319.25	188	260.57	79	314.28	302	298.09	569	13.432
3	313.15	253	259.44	96	313.73	295	295.44	644	11.679
4	437.38	723	339.03	209	430.96	663	402.45	1,595	17.794
5	432.87	901	336.69	191	427.97	682	399.18	1,674	15.901
6	442.69	719	343.40	238	435.60	768	407.23	1,725	18.129
7	539.66	1,285	418.32	361	529.22	1,317	495.73	2,963	19.212
8	530.49	1,266	409.30	321	518.54	1,279	486.11	2,866	21.543
9	527.72	1,266	406.91	338	516.68	1,162	483.77	2,766	19.328
10	631.93	2,310	508.30	901	566.13	1,323	568.75	4,534	24.484
11	630.67	2,268	509.98	938	563.97	1,355	568.21	4,561	22.039
12	635.92	2,311	513.50	974	569.58	1,455	573.00	4,740	24.544
13	675.77	2,457	555.27	1,377	599.97	1,598	609.67	5,432	23.416
14	687.21	2,980	565.44	1,415	612.73	1,679	621.73	6,074	25.987
15	682.81	2,924	561.13	1,351	608.31	1,806	617.41	6,081	23.750
17	739.60	3,215	713.38	3,450	734.44	3,031	729.14	9,696	29.308
18	736.21	3,179	712.64	3,515	736.20	3,271	728.32	9,965	27.483
19	734.85	3,144	708.10	3,360	731.48	3,151	724.81	9,655	28.745
20	728.93	3,091	693.37	3,151	726.81	3,026	716.37	9,268	25.978
21	724.84	3,162	692.41	3,153	725.34	3,065	714.34	9,380	29.928
22	725.98	3,038	691.31	3,110	726.11	3,068	714.47	9,216	25.948
23	803.65	3,853	760.75	4,105	810.95	3,868	791.79	11,826	31.394
24	807.73	3,760	760.19	4,099	814.33	3,737	794.08	11,596	28.326
25	814.26	3,881	762.36	3,912	819.33	3,930	798.65	11,723	31.712
26	894.43	4,884	838.64	5,109	904.62	4,750	887.90	14,743	30.84
27	902.31	4,824	846.03	5,065	908.85	4,753	885.54	14,642	33.49
28	901.69	4,820	845.58	4,970	910.11	4,733	885.79	14,523	30.53
29	901.77	4,807	845.84	5,200	909.03	4,639	885.73	14,645	33.65
30	904.52	4,836	847.52	5,083	911.64	4,728	879.23	14,647	29.98

From the results of these trials it was estimated that the mean of the revolutions of the three screws for the various speeds required on the four-hour trials was as follows:

At 12 knots, 283 revolutions per minute.
At 16 knots, 377 revolutions per minute.
At 20 knots, 477 revolutions per minute.
At 25 knots, 624.5 revolutions per minute.
At 27½ knots, 701 revolutions per minute.
At 29½ knots, 776.5 revolutions per minute.

Temporary tanks, erected on the deck of the vessel, were employed in measuring the amount of fuel oil and water used during the various four-hour trials. These tanks were carefully calibrated and the data on fuel oil and water consumption may be taken as accurate, except that no account is taken of the amount of water pumped in from the reserve tanks for making up losses.

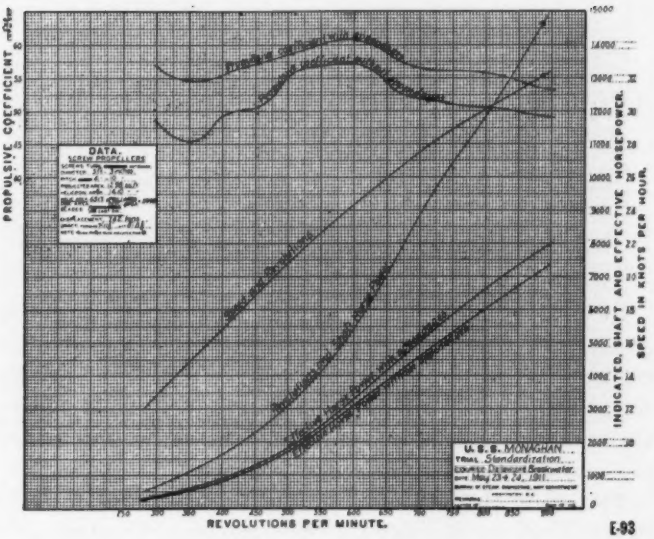
The various four-hour trials were carried out between May 24th and 27th, 1911. A general synopsis of the results is given in the following table. The usual maneuvering trials were carried out with very satisfactory results.

DATA FOR FOUR-HOUR RUNS.

Date and time of trial..	May 25, A. M.	May 25, P. M.	May 26, P. M.	May 26, A. M.	May 24, P. M.	May 27.
Mean draught on trial, feet and inches.....	8–05½	8-04½	8–04½	8–04½	8–03⅞	8–04
Correspond'g displacement, tons.....	747.5	740.3	736.2	736.2	732.6	734.8
Trim by stern, inches...	11¼	11¾	15	14	13	16½
Speed of ship, knots per hour.....	11.933	16.119	20.121	25.318	27.755	30.449
State of sea.....	smooth long swell	smooth long swell	smooth	smooth	short chop	short mod.
Revolutions of main shafts:						
Starboard	290.278	397.94	522.480	700.597	721.706	833.03
Center.....	268.958	350.94	409.850	572.384	686.300	778.71
Port	285 433	390.57	508.750	628.294	720.146	837.97
Mean revolutions.....	281.556	379.93	480.360	633.758	709.383	816.56
Slip of propellers in per cent. of their speed:						
Starboard	13.81	15.07	19.26	24.23	19.36	23.36
Center.....	6.98	3.70	—2.93	7.26	15.22	18.02
Port.....	12.35	13.47	17.08	15.51	19.18	23.82
Mean slip	11.05	10.75	11.14	15.67	17.95	21.73

Steam pressures, per gage:	May 25, A. M.	May 25, P. M.	May 26, P. M.	May 26, A. M.	May 24, P. M.	May 27,
Boilers	237	240	258	259	253	235
Main steam pipe at throttle	237	240	258	259	252	233
Auxiliary exhaust	6.9	15.4	13.9	17	12.8	16.7
On glands	1.2	1.6	1.72	1.7	1.3	1.5
Pressures, absolute:						
Main H.P. steam chest	15	25.8	39.7	94	159	202
H.P. cruising steam chest	55.5	108.7	239.6	9	15.5	9
I.P. cruising steam chest	28.7	50.5	109.8	25.9	13.3	5
L.P. turbine chest, Starboard	6.5	11.2	15.4	25.2	37.9	49
Port	7.1	10.7	15.0	25.6	38.6	50
Pressure in lubrication system, gage	60	60	44.6	60	60	70
Vacuum in condenser, inches, starboard	29.1	29.6	30	29.97	29.5	28.59
port	28.9	29.5	29.7	29.30	28.2	28.18
Air pressure in fire-room, ins. of water	1.1	2.1	1.18	1.6	2.46	3.87
Barometer	29.80	29.82	29.88	29.89	29.92	29 90
Auxiliaries:						
Revolutions or double strokes:						
Main air pump, Starboard	53.8	60.5	34	42	43.7	50.4
Port	...	...	33.8	42	44	50.9
Main circ. pump, Starboard	266	256	250	293	303.1	294
Port	264	252	250	285	279	292.5
Main feed pump, Forward	8.9	11.8	...	...	18.8	28.8
Aft.	...	...	14.8	30.5	25.3	28.8
Fuel-oil serv. pump, Forward	...	...	...	38.2	44.5	56.8
Aft	17.2	21	28.3	...	40	45
Lub. oil serv. pump, Starboard	80	80	35.2	43.3	64.3	53
Port	...	...	35.2	43.3	44	53
Fire and bilge pump.	33.7	37.8	36.7	41.8	38	37.7
Forc.-draft blowers	1,102	1,020.5	1,534	1,390	1,093	1,426
Dyn. engines, watts	2,975	3,630	3,160	2,730	3,208	3,308
Number of boilers in use	1	1	2	4	4	4
Number of burners in use (each boiler)	8	10	9.7	9	11	11
Press. of oil to burners.	200	200	200	200	201	197
Temperatures:						
Oil to burners	184	160.4	161	160	180.2	106.2
Injection	59.6	59	59.4	58	58.5	57.7
Discharge, starb'd	64.1	65.6	71.0	76.6	84.6	93.3
port	64.3	66.2	70.7	81	89.9	94.4
Air-pump discharge, starboard	...	...	71.8	75.5	89.1	91.1
port	69.5	67.7	71.8	75.5	89.1	91.1

	May 25, A. M.	May 25, P. M.	May 26, P. M.	May 26, A. M.	May 24, P. M.	May 27.
Temperatures:						
Feed	132.5	103.4	112	102	102 6	197
Outer air	71.3	66.4	64	60	59.7	63
Engine-room working level	108	100.5	99.7	88	85.8	96
Fireroom working level	103.5	100.5	102	100	91.4	95
Horsepower, main shafting:						
Starboard	181	531	1,234	3,267	3,177	4,131
Center	146	317	378	1.461	2,989	4,191
Port	185	489	1,088	1,900	3,073	4,088
Total	512	1,337	2,700	6,628	9,239	12.410
Water:						
Pounds per hour, all machinery	22,045	32,082	48,564	99,904	149,680	192,261
Pounds per hour, per H.P. of main shafts, all machinery	43.057	23.996	17.987	15.073	16.201	15.492
Fuel oil:						
Analysis and heating power	C, 86.04; H, 13.25; S, 0.34; B.T.U., 19,390.					
Pounds per hour	1,942.5	2,957	4,355.5	8,842.5	12,266	16,497
Pounds per knot run	162.78	183.45	216.36	349.26	441.94	541.79
Knots per ton	13.361	12.211	10.348	6.414	5.069	4.134
Deduced data:						
Square feet of H.S. per S.H.P.	8.789	3.366	3.333	2.716	1.948	1.450
Pounds of fuel oil per S.H.P. per hour	3.794	2.212	1.613	1.334	1.328	1.329
Pounds of fuel oil per burner per hour	242.81	295.70	224.51	245.62	278.77	374.93
Pounds of water evaporated per pound of oil	11.349	10.850	11.150	11.298	12.203	11.654
Square feet of cooling surface in condensers per S.H.P.	18.75	7.180	3.556	1.448	1.039	.774
Auxiliaries in operation:						
Main air pumps	1	1	2	2	2	2
circ. pumps	2	2	2	2	2	2
feed pumps	1	1	1	1	2	2
Forced lubrication pumps	2	2	2	2	3	3
Oil-cooling pump	...	...	...	...	1	1
Fire and bilge pumps	1	1	1	1	1	1
Fuel-oil service pumps	1	1	1	1	2	2
Oil-tank pump	One as required.					
Dynamos	1	1	1	1	1	1
Forced-draft blowers	1	2	1	2	4	4
Steering engine	As required.					
Feed heaters	1	1	1	1	1	1
Steam to galley	As required.					


PROPULSIVE COEFFICIENT.
INDICATED, SHAFT AND EFFECTIVE HORSEPOWER.
SPEED IN KNOTS PER HOUR.
REVOLUTIONS PER MINUTE.
DATA.
SCREW PROPELLERS
U. S. S. MONAGHAN.
TRIAL Standardization.
COURSE Delaware Breakwater.
DATE May 23-24, 1911.
E-93

A PAGE IN THE HISTORY OF AMERICAN TORPEDO PROGRESS.

By G. W. Baird, Rear Admiral (formerly Chief Engineer), U. S. N.

In 1874–5, when the writer was on duty in the Bureau of Steam Engineering, as assistant to Engineer-in-chief Wood, the latter began the preparation of some papers on torpedoes for a Committee of Congress, and the writer was detailed to aid in the work. The papers were, however, never presented to Congress, nor were they ever completed.

The writer obtained from the late Mrs. Colt, widow of Col. Samuel Colt, the sketches and the fragments of reports which the Colonel had used, many of which have never before been made public.

The sketches herein reproduced were copies from the originals, as nearly as the writer and Second Assistant Engineer (now Real Admiral) W. B. Bayley, could imitate them.

Without a doubt the credit for the first submarine cable belongs to Colonel Colt, and its success is proved by the following reports.

Plate VII shows a plan for determining a ship's position by means of two operators, which seems to have originated with Colonel Colt, and which has been so successfully practiced in later warfare. The observers were placed at A and B (see plate), each having lines of observation, and were able to communicate with each other over an independent wire AB.

The writer was not able to determine whether or not the explosion was effected by the circuit being completed by one or by both the observers, but that is of no importance so long as the communication between the observers was not interrupted.

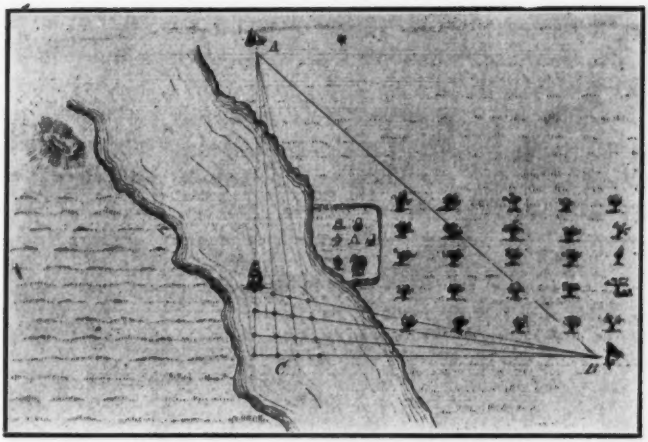


Plate VII.

Plate VIII is a general plan of Colt's system of torpedoes for operation by a single observer, which the writer believes has never been successfully developed by any person other than Colt.

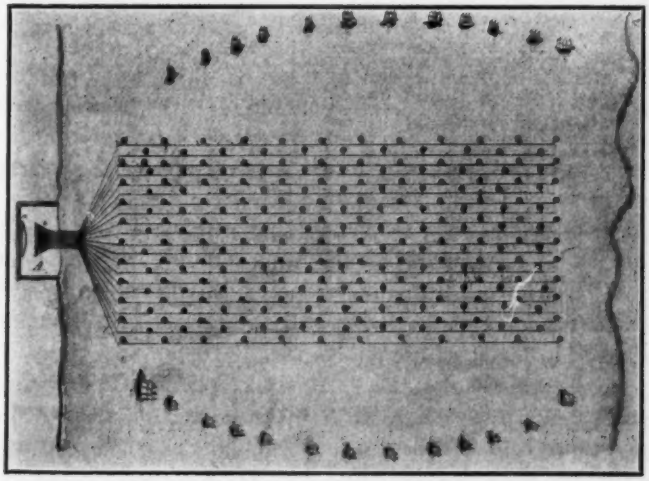


Plate VIII.

The observer was seated in a bomb-proof structure at A, having at his back a convex mirror (a), and in front of him a keyboard (b).

Each torpedo was connected to a key, on the board, and when the circuit was closed between that particular torpedo and its corresponding key the explosion of that individual torpedo occurred.

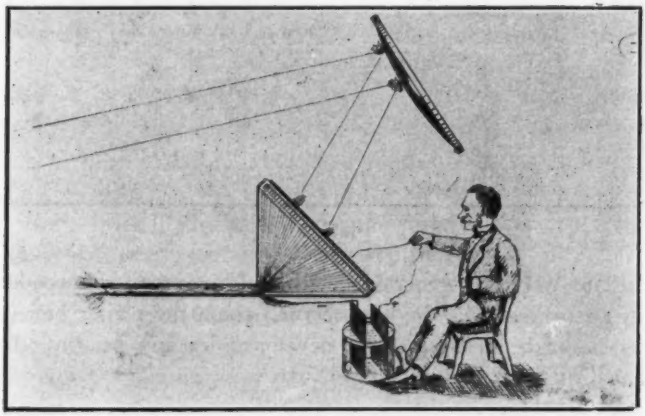


Plate IX.

Plate IX shows the elevation of the observer inside the bomb-proof. It will be seen that the observer has one wire in his hand and that there are two sets of wires entering the observatory; one set going to the negative pole of the battery and the other set attached to the keys.

The torpedoes were so planted that when an object should float upon the surface of the water immediately above a torpedo, the image of that object would be reflected by the convex mirror upon the keyboard indicating which circuit should be made to that particular torpedo.

The writer was not able to ascertain whether Colonel Colt used the electric spark to ignite his explosives or not, but the appearance of a sketch of a lock (fig. 16), which was found among the drawings, indicated that this was used instead.

Mr. Colt, like his predecessors, met with great opposition

from Government officials, as the correspondence of that period will show. The Navy was at that time controlled by the old Board of Commissioners, who were much wedded to sails for propulsion and were averse to innovations. Such expressions as "cowardly, and no fair nor honest warfare" were currently reported as being uttered by officials high in rank.

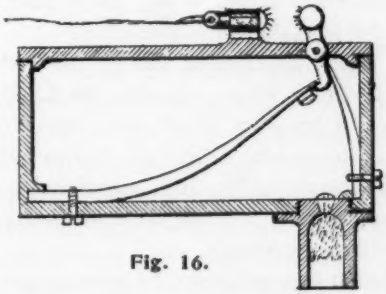


Fig. 16.

But Congress, in its wisdom, appropriated $17,000 for the prosecution of Colt's experiments, and the Commandant of the Navy Yard was authorized by the Secretary of the Navy to give him certain assistance.

The following letter of Col. Colt will complete this sketch of torpedo history in the U. S. Navy.

"WASHINGTON, D. C., *22d April, 1844.*

"To the HON. JOHN T. MASON,

"*Secretary of the Navy.*

"Sir: I have the honor to report the completion of the experiments with my submarine battery, made under the instruction of your Department in conformity with the Joint Resolution of Congress approved 30th April, 1842.

"Previous to the authority of Congress to test my invention I made a public experiment in the harbor of New York on the 4th day of July, 1841, on a vessel of about 100 tons burden; this vessel, while floating at the rate of about three miles an hour by the tide, and in tow of a boat from the U. S. Ship *North Carolina,* was blown to atoms. The boat

having her in tow was at a distance of less than 100 feet from the exploded vessel, and escaped without the least injury to herself, as did all the other objects not intended to be destroyed. My position in this case was about a quarter of a mile distant from the vessel.

"The result of this experiment proved that by means of my submarine battery an enemy's vessel could be effectively destroyed while underweigh, within pistol shot of our own ships without the least danger to ourselves.

"The next public experiment was on the 30th of August, 1842, in the Potomac River, opposite the U. S. Arsenal at Washington, on a schooner of 60 tons. On this occasion I operated from Alexandria, a distance of five miles, and blew the vessel to pieces so quickly after the signal was given that the noise of my explosion and the noise of the signal gun were blended together in one sound.

"This explosion was witnessed by most of the members of Congress and by the executive officers of the Government at Washington, and proved conclusively that my battery could be brought to bear without loss of time upon an enemy at a distance far beyond the reach of guns of the largest caliber.

"Another experiment was made in the harbor of New York, opposite the Battery, on the 18th of October, 1842, before a committee of the American Institute. The object destroyed was the brig *Lotta,* of 260 tons burden. She was stationed, and at the distance of three-fourths of a mile from where I was stationed, I being on board the Revenue Cutter *Ewing,* near Governor's Island. The vessel in this case was more than four times as large as that used upon the last occasion, and as proof of the entire success of the experiment I beg leave to subjoin a copy of a resolution of the American Institute on this subject

"'AMERICAN INSTITUTE,

"'NEW YORK, *20th October, 1842.*

"'*Extract from the minutes.*

"'"*Resolved,* that the thanks of this committee are due and are hereby tendered to Mr. Samuel Colt for the grat-

uitous exhibition of his submarine battery in successfully blowing up a vessel of 260 tons, which enabled this committee to give a thorough test of the efficiency of the battery, and also to try its effect on life boats, life preservers, etc., which had been offered for competition.

"'"*Resolved,* that a copy of the above resolutions, signed by the chairman and secretary, be forwarded to Mr. Colt.

"'GEO. C. DEKAY,

"'*Chairman, Nautical Committee.*'

"A fourth public experiment took place on the 13th instant in the Eastern Branch of the Potomac between the Arsenal and the Navy Yard, upon a ship of 500 tons burden, and under full sail with a moderate breeze and favorable tide.

"My position in this case was upon the opposite bank of the river, something more than two miles distant from the ship; the power used in this instance was as small as possible to produce the desired effect. The explosion struck the ship immediately under the foremast and her whole hull forward of the main mast was blown into fragments, and the remaining portion of the wreck descended into the bed of the river.

"In preparing for this experiment the channel was so fortified that the largest fleet which might have entered the Branch could have been as easily and effectually destroyed in the same space of time as was the vessel which was the subject of this exhibition. The explosions which preceded that made upon this ship were for the purpose of showing the certainty with which an enemy's vessel could be destroyed without the least injury to our own.

"The number of small boats upon the river, which, from the position I occupied, could not be distinguished from pieces of the wreck, deterred me from proceeding further after the destruction of the vessel, for fear of accident.

"On the 30th instant I made another experiment upon the sunken wreck which was blown up at the signal with a charge from my battery; also several other separate explosions in

the water in the vicinity of the wreck, either of which was sufficient to destroy any vessel which could enter the channel. On this occasion the signal was given from the Navy Yard, and the explosion was made and the whole wreck blown into the air before the report of the signal gun had ceased. By those persons who were on the shore opposite and near by, the object was to show how quickly an explosion could be made after the signal was given.

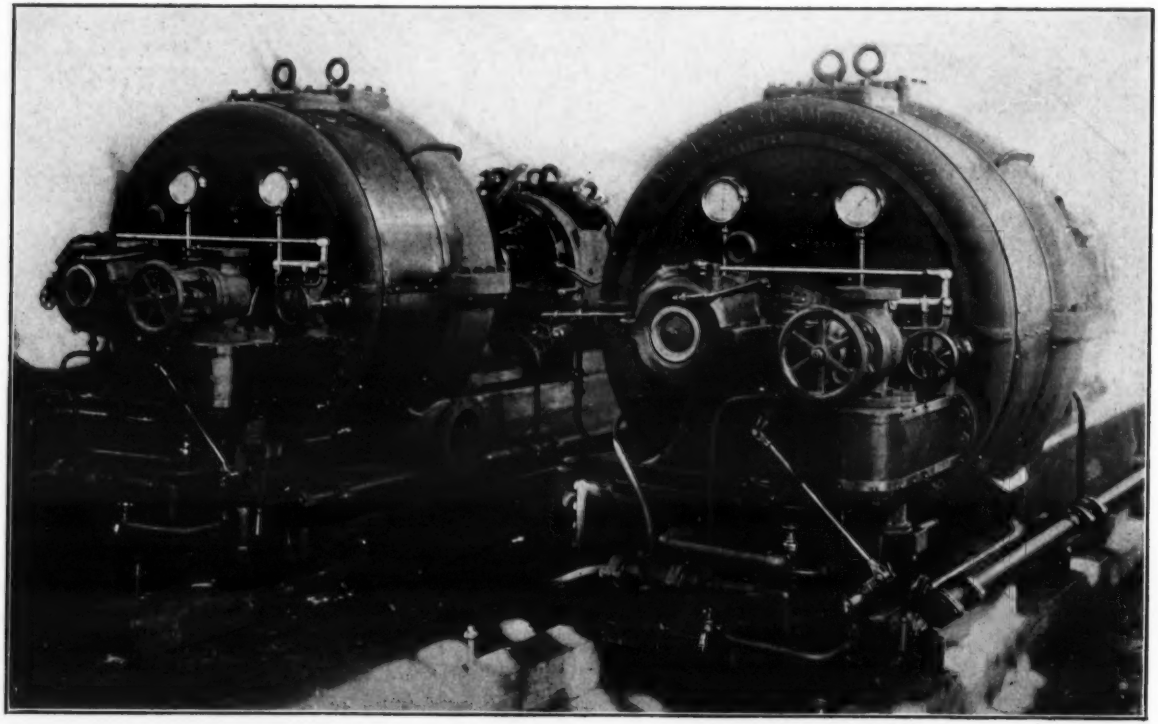
"In submitting this report I have the satisfaction to state that in no single instance have I failed in the use of my submarine battery, and that the combined results of my experiments prove beyond all doubt that it can be successfully employed with the most perfect ease and safety to destroy the largest class of ships of war when in motion passing in or out of harbor, without the necessity of approaching within reach of shot from the guns of the largest caliber; that continued operations of the destruction of any number of vessels can be effected without uncovering the means under exposure of an advancing squadron, and that it can be used for the defense of any harbor, without endangering the passage, in or out, of other than hostile vessels.

"I have the honor to be,

"Very respectfully, Sir,

"Your obedient servant,

"SAM'L COLT."



TERRY TURBINE AND GENERATOR, U. S. S. "YOSEMITE."

TESTS OF TURBO-GENERATORS FOR U. S. S. *YOSEMITE.*

By Chas. A. Gleason, Assistant Inspector of Engineering Material, Navy Department.

Two variable-voltage, battery-charging and lighting turbo-generators for the U. S. S. *Yosemite,* formerly the U. S. Army Transport *Ingalls,* were tested at the plant of the Terry Steam Turbine Company, Hartford, Conn., June 29th and 30th and July 3d, 11th and 14th, 1911.

DESCRIPTION OF TURBINES.

The turbines are of the Terry two-stage impulse type, expanding steam in the first stage practically to atmospheric pressure. The second or low-pressure stage takes care of the expansion from atmosphere to condenser pressure. Working condition 150 pounds steam pressure and 27-inch vacuum. Steam is admitted to a steam chest in which are a strainer and emergency spring-loaded valve and the main governor valve, the steam passing from the pipe to the turbine in the above order. The main governor is fitted directly to the turbine shaft and is directly connected to a double-seated balanced valve. The emergency governor is fitted to a gear-driven shaft which also drives the pump for the lubrication system. The pump is of the plain-geared type without valves, taking oil from a reservoir in the base. The oil is maintained at a constant pressure by means of relief valve. The turbine casing and main bearings are carried by two pedestals which eliminate any trouble that might be caused by the heating of the casing. There is no end thrust due to the action of the steam, but a suitable thrust block running in oil is pro-

vided to take care of any thrust that might be caused by the movement of the ship. The turbine is directly connected to the generator by means of a coupling, flexibility being obtained at this point by means of steel pins and rubber bushings. Generator and turbine are mounted on the same cast-iron base plate.

DESCRIPTION OF GENERATORS.

The generators were built by the Diehl Manufacturing Co. of Elizabethport, N. J., and are of the type "K," which this company has designed for the Navy with a view to minimizing weight and floor space.

The yokes are steel, and circular in shape. The main poles, four in number, are sheet-steel punchings, riveted together and held in place by two pole bolts. The interpoles are of machine steel and wound with bare strap copper. The winding is spread over the pole length in a manner calculated to give a good ventilating effect.

The coils are held in the slots by a number of phosphor-bronze bands placed on the core and by heavy steel end bands.

The armature is of the double-commutator type, one-half of the load coming from each side.

The commutator bars are of "V" construction and are protected against centrifugal strains by two heavy steel shrink rings.

The armature core and commutators are mounted on a shaft connected to the turbine by means of the above-mentioned flexible coupling.

The brush rigging is rigidly supported from the generator frame.

These machines are for battery charging and are given a nominal rating of 85 kw., 125 volts, 2,300 r.p.m. They generate the full 85-kw. capacity at any voltage from 110 volts to 175 volts, and are compounded so as to give maximum variation of 3.8 volts from full load to no load and back to full load in 100-ampère steps.

The brush holders are of the box type and are designed to give a positive " fast" action.

METHOD OF CONDUCTING THE TEST.

The method of conducting the test was as follows: Steam pressure by gage, temperature and calorimeter measurements were taken immediately before the throttle valve of turbine. The exhaust pressure was measured by mercury column placed directly at the turbine exhaust. The exhaust steam was taken to a surface condenser, the hot-well pump of which discharged into two tanks mounted on carefully-adjusted Fairbanks standard weighing machines. In this way the weight of the condensed steam was measured.

The condenser was tested out for leakage before and after the test and this leakage was found to be negligible. The temperature of the bearings was taken by placing a thermometer in the circulating oil directly after it left the bearing.

The load was absorbed by a water rheostat and was measured by Weston standard volt and ammeters. Throughout the heat run temperatures were taken by thermometers at stated intervals, and at the end of the heat run such temperatures were obtained of the revolving parts as could not be recorded during operation.

The first test conducted was that for speed regulation, followed by the compounding test, then a four-hour full-load heat run. Immediately after the completion of the full-load test the machine was shut down and temperatures taken. Between the completion of the full-load test and the commencement of the overload run no time was allowed to elapse except that required for obtaining temperatures. After an overload test of two hours final temperatures were recorded.

Following the overload run the machine was tested out under full load at voltages from 100 to 175, the operation being satisfactory in each case. The emergency governor tripped at 2,480 r.p.m., or about 18 per cent. above the rated speed and 10 per cent. above average heat-run speed. At the

conclusion of these tests the machine was run at half load to determine the water rate at this point, the water log being kept for 30 minutes.

During the test the machines were mounted only on skids without holding-down bolts and operated without undue vibration or noise.

Plates 1 and 2 show the general arrangement and special features of the machines which are further illustrated in the accompanying photograph.

Plate 3 contains curves giving the water consumption at various loads, plotted on the assumption that the total steam follows the straight-line law.

Plates 4 to 13 give in tabulated form the results of the various tests which are summarized in Plate 14.

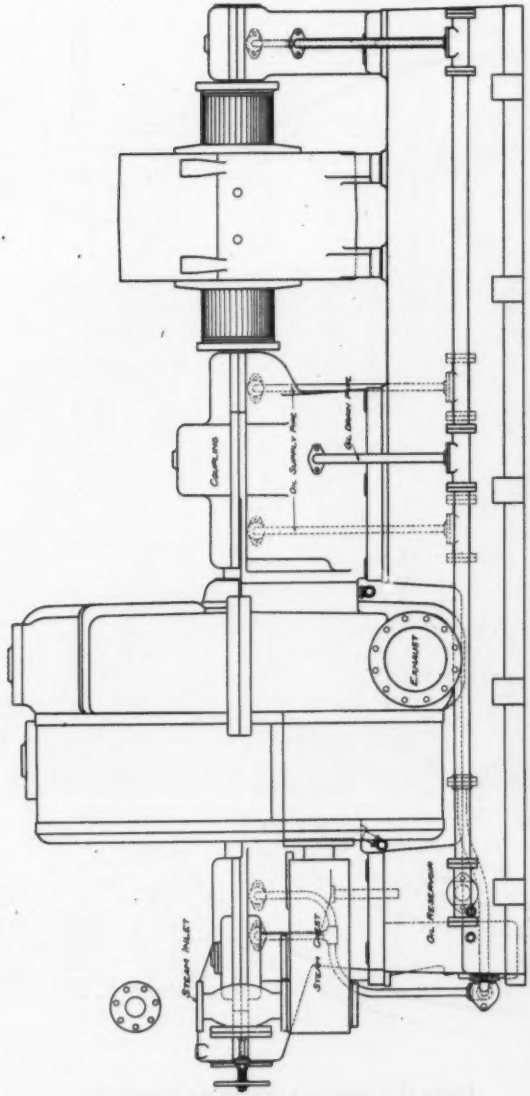


Plate I.—SIDE ELEVATION OF TURBINE AND GENERATOR.

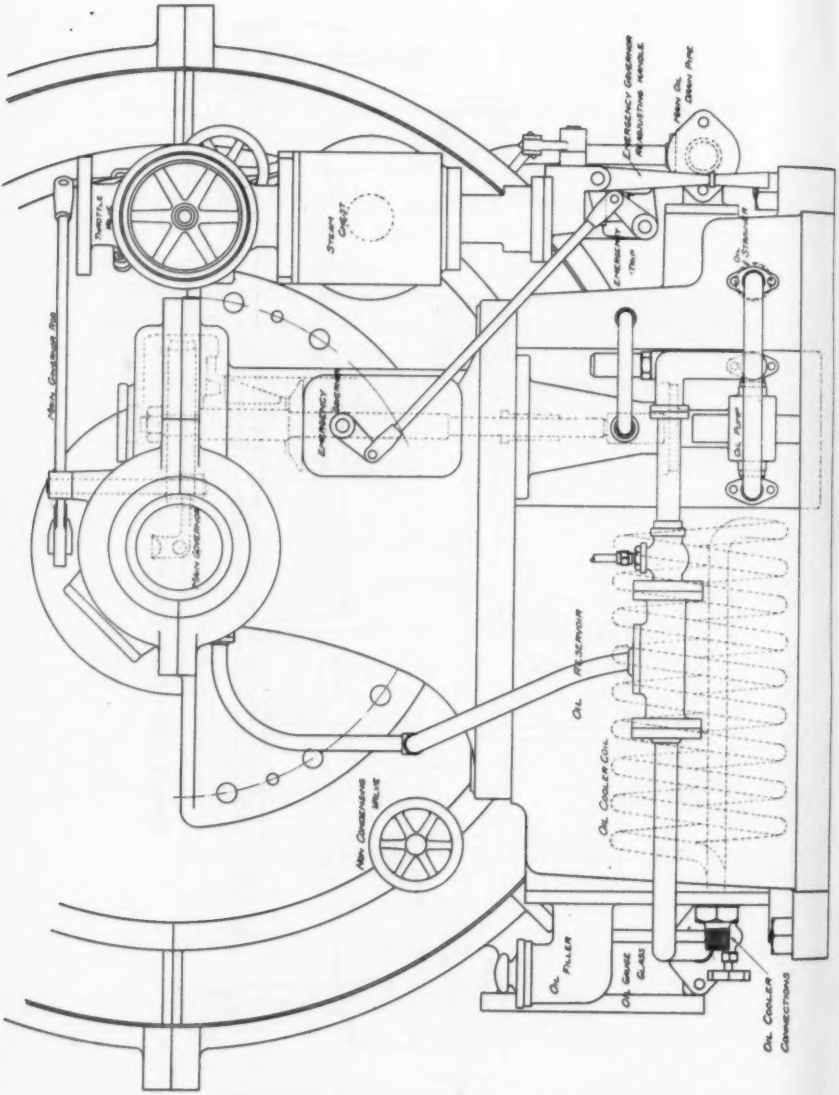


Plate II.—END ELEVATION OF TURBINES.

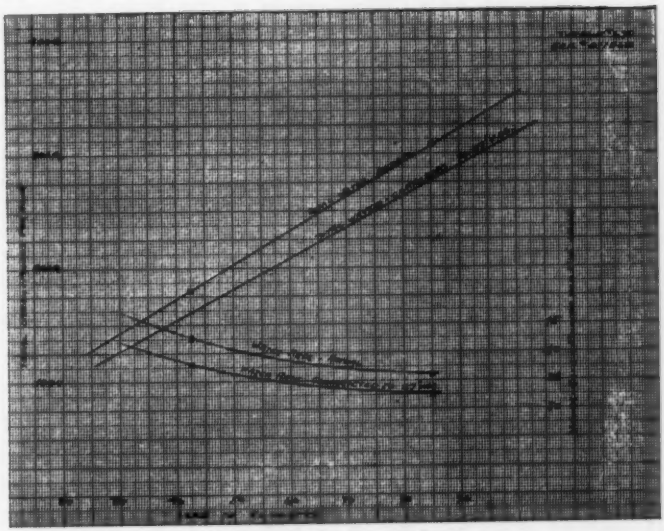


Plate III.

TURBINE # 670
GEN. # 27/940

HARTFORD, CONN. June 29, 1911.

COMPOUNDING TEST.

ARMATURE VOLTS	ARMATURE AMPERES	R. P. M.	STEAM.	VACUUM			
130.	680	2230	150	23.5			
131.5	570	2240	"	23.75			
128.2	420	2260	"	24.			
131.	325	2260	"	24.25			
131.	225	2265	"	24.5			
129.5	130	2270	"	24.5			
131.	0	2280	"	24.8			
128.8	100	2270	"	24.8	Maximum Voltage Minimum Voltage		131.5 128.2
129.2	200	2260	"	25.0	Difference		3.3
131.	300	2260	"	24.8			
130.5	400	2245	"	24.9			
131.5	500	2230	"	24.5			
129.	600	2180	"	24.2			
128.5	680	2205	"	23.7			
130.	0	2265	"	25.2			
130.	680	2220	"	23.5			

Plate IV.

TURBINE #670
GEN. #271960

ENGINE REGULATION

HARTFORD, CONN. JUNE 28, 1911.

STEAM	WC	AMP	REX	VOLTS	JUMP REVOLUTION HIGH	LOW	STEADY	CHANGE	% CHANGE STEADY	% CHANGE MOM.	JUMP VOLTS HIGH	LOW	STEADY	JUMP	% JUMP	
		90% FULL LOAD TO NO LOAD NONCONDENSING 90% STEAM PRESSURE														
180	1.0	612	2250	121.												A
180	1.4	0			2290		2260	30	1.33	1.78	134.5		133.	13.6	11.2	
180	1.0	612				2240	2250	30	1.32	1.75		120.	122	13	9.8	
		FULL LOAD TO NO LOAD CONDENSING 60% STEAM PRESSURE														
180	22.5	680	2.230	128.												
180	22.5	0			2275		2266	45	2.03	2.45	139.6		138.	11.5	9.0	B
180	22.5	680				2200	2215	50	2.2	2.87		118.	128.5	14.	10.8	
		FULL LOAD TO NO LOAD NONCONDENSING 100% STEAM PRESSURE														
150	.8	680	2250	124.5												
150	1.2	0			2300		2270	30	1.33	2.22	141		137	16.5	13.2	C
150	.7	680				2230	2240	40	1.76	2.19		120	123.5	17.	12.4	
		FULL LOAD TO NO LOAD CONDENSING 100% STEAM PRESSURE														
150	22.5	680	2240	124.5												
150	21.5	0			2300		2260	20	.85	2.68	136		117.5	11.5	9.2	D
150	21.5	680				2220	2230	40	1.77	1.77		109	128.7	8.5	7.2	
		FULL LOAD TO NO LOAD NONCONDENSING 100% STEAM PRESSURE														
180	.75	680	2250	128.												
180	1.3	0			2280		2270	20	.89	1.33	193.5		191.5	19.5	11.2	E
180	.75	680				2234	2290	30	1.32	1.76		125.5	129.	16.	11.3	
		FULL LOAD TO NO LOAD CONDENSING 120% STEAM PRESSURE														
187	22.6	680	2245	125.												
192	23.5	0			2300		2269	20	.89	2.44	134		119.	9.	7.2	F
190	22.	680				2210.	2245	20	.88	2.43		103	129	11.	9.2	
		FULL LOAD TO 80% OF FULL LOAD CONDENSING 100% STEAM PRESSURE														
150	22.	680.	2215	123.												G, GRADUAL
150	22.4	136			2270		2265	50	2.25	2.48	130		125.4	7.	5.7	
150	22.7	680.				2180	2215	90	1.77	3.75		115	123.	10.5	8.4	
150	22.7	680	2220	129.												G_2 ONE STEP
150	22.5	136			2270		2260	90	1.8	2.25	132		125.	8.	6.0	
150	22.7	680				2130	2240	80	.86	3.54		114	125.5	11.	8.8	

Plate V.

Turbine #670
Gen. #271960

Hartford, Conn. June 30, 1911.

Temperatures by Thermometer

	Cold	End of Full Load Run		End of 33⅓% Overload Run		
		Temp.	Rise	Temp.	Rise	
Air at Machine	23.	28.5	5.5	30.	7	
Armature Surface	23.	51.	22.5	65.	35.	
Armature Core	23	51.	22.5	68.	38.	
Commutator Bars	23.	42.8	14.3	50.5	20.5	
Pole Tip Leading	23.	51.	22.5	65.0	35.	
Series Field	23.	49.	20.5	59.5	29.5	
Shunt Field	23	49.	20.5	59.5	29.5	
Series Shunt	23.					
Armature Coil	23.	48.	19.5	60.	30.	
Field Rheostat	23.	47	18.5	45.	15.	
Inboard Bearing	23.	48.5	20.	48.5	18.5	
Outboard Bearing	23	45.5	17.	45.5	15.5	
Interpole	23	50.	21.5	66.	36.	

Full Load at Various Voltages.

Volts	Amp.	
100	850	
110	773	Emergency Trip operates at 2480
120	707	10.7% overspeed
130	654	
140	606	
150	566	
160	530	
175	485	

Operation timed at these voltages and found to be O.K.

Plate VI.

TURBINE #670
GEN. #271360

HARTFORD, Conn. JUNE 30, 1911.

(ONE HOUR) FULL LOAD WATER RATE - (TAKEN DURING 4-HOUR HEAT-RUN)

TIME	TERMINAL VOLTS	TERMINAL AMP.	STEAM	VACUUM	R.P.M.	K.W.	TEMP. CAL. DEG. F.	WATER READINGS NET.	QUALITY	WATER RATE ACTUAL PER K.W. HOUR.	WATER RATE CORR. TO DRY STEAM AT 27" VAC.
11.15	126.	680.	150.	23.9	2245	85.7	328.				
21	125	680	150	23.8	2235	85.	324.	308.			
27	125	680	150	23.9	2240	85.	328.	309.			
33	125	680	150	23.9	2250	85.	329.	309.			
39	125	680	150	24.	2255	85.	337.	3095			
45	125	680	150	23.9	2260	85.	326.	3085			
51	125	680	150	23.9	2250	85.	327.	304.			
57	125	680	150	24.	2230	85	329.	313.			
12.03	125	680	150	24.	2235	85	332.	303.			
.09	125	680	150	24.	2215	86	330.	309.			
.15	124.	680	150	24	2230	84.4	328.	308.			
AVGE.	125	680	150	23.9	2240	85.	329.	3090. PER HOUR	10° S.H.	36.25	32.6
(HALF-HOUR) - HALF LOAD WATER-RATE (TAKEN AFTER HEAT-RUNS)											
5.52	126	340	150	24.2	2260	42.5	319.				
5.57	125	340	150	24.5	2255	42.5	321.	333.5			
6.02	125	340	150	24.7	2260	42.5	321.				
6.07	125	340	150	24.7	2270	42.5	321.	270.			
6.12	125	340	150	24.8	2270	42.5	319.				
6.17	125	340	150	24.5	2270	42.5	321.	302.			
6.22	125	340	150	24.7	2270	42.5	321.				
AVGE.	125	340	150	24.6	2265	42.5	320.4	905.5 PER HALF HOUR		42.6	38.
							1811. PER HOUR				

Plate VII.

TURBINE #670
GEN. #271960

HARTFORD, CONN. JUNE 30, 1911.

4 HOUR – FULL LOAD HEAT RUN.

TIME	TERMINAL		STEAM	VACUUM	R.P.M.	SHUNT FIELD			TEMPERATURE		
	VOLTS	AMP				VOLTS	AMP	RES.	AIR	RHEO.	SH. FIELD
9.45	126	680	150	24.5	2250	73.	2.27	32.1	23.5	23.	23.
10.00	125	680	150	24.4	2250	73.5	2.25	32.6	24.2	28.3	
15	125	690	150	24.1	2245	74.	2.24	33.	24.6	31.5	36.5
30	125	680	150	23.7	2230	73.5	2.17	33.8	25.0	34.	38.
45	125	680	150	23.6	2230	73.	2.14	34.1	25.5	34.	39.
11.00	125	680	150	24.0	2230	75.	2.16	34.7	26.	35.	40.
15	126	680	150	23.9	2245	76.5	2.18	35.1	26.	36.	41.
30	125	680	150	23.9	2230	73.5	2.07	36.5	26.8	37.	41.5
45	126	680	150	23.9	2260	76.5	2.15	35.6	27.	44.	42.
12.00	125	680	150	23.9	2220	80.	2.23	35.9	28.	45.	43.
15	124	680	150	24.	2230	76.	2.12	35.8	27.5	46.5	43.
30	125	680	150	23.4	2210	76.5	2.12	36.1	28.	47.	43.
45	125	680	150	23.	2250	75.	2.07	36.2			
1.00	125	680	150	22.9	2230	77.5	2.13	36.3	28.5	47.	43.5
15	125	680	150	22.8	2235	79.	2.17	36.4	28.5	48.	43.8
30	125	680	150	22.5	2235	77.5	2.13	36.4	29.5	49.	44.
45	125	680	140	22.6	2110	87.	2.38	36.6	29.	47.	45
							Commutation 3/4				

2 HOUR RUN AT 33⅓% OVERLOAD.

2.35	125	920	150	21.5	2190	75.	2.10	35.7	29.	34.	45.
50	125	910	150	21.	2190	76.5	2.12	35.6	29.	37.5	47.
3.05	125	910	150	21.	2180	77.	2.13	36.2	29.	39.5	49.
20	125	910	150	21.2	2180	78.5	2.14	36.7	29.	41.	49.
35	125	910	150	21.2	2180	79.	2.13	37.	29.	43.	51.
50	125	920	150	21.2	2200	81.	2.17	37.3	30.	44.	51.
4.05	125	915	150	20.9	2185	83.	2.22	37.3	29.5	44.	51.5
20	125	910	150	21.3	2180	82.5	2.16	37.8	30.	45.	51.5
35	125	925	150	21.3	2200	83.	2.18	38.1	30.	45.	51.2
							Commutation 3/4				

Plate VIII.

TURBINE # 671
GEN # 271861

HARTFORD, CONN, JULY 3, 1911.

ENGINE REGULATION

STEAM	WORM	AMP	REV	VOLTS	JUMP REVOLUTION HIGH	LOW	STEADY	CONNECTING STRAIT	CONNECTING PAR.	JUMP VOLTS HIGH	LOW	STEADY	JUMP	% JUMP	
		90% FULL LOAD TO NO LOAD NONCONDENSING 60% STEAM PRESSURE													
120	1.75	612	2220	124.5											A
120	2.7	0			2280		2265	45	2.03	2.70	132		116.5	7.5	6.	
120	1.7	612				2210	2235	30	1.32	2.43		110	123.5	6.5	5.5	
		FULL LOAD TO NO LOAD CONDENSING 80% STEAM PRESSURE													
120	2.75	680	2265	121											B
120	2.25	0			2330		2300	35	1.54	2.87	131.		122.5	10.	8.3	
120	2.1.	680				2240	2250	50	2.17	2.61		113.	122	10.5	8.75	
		FULL LOAD TO NO LOAD NONCONDENSING 100% STEAM PRESSURE													
150	1.5	680	2245	124.											D
150	2.5	0			2290		2275	45	2.0	2.0	134		121.	10.	8.3	
150	1.6	680				2220	2245	30	1.3	2.42		111.	124.	10.	8.26	
		FULL LOAD TO NO LOAD CONDENSING 100% STEAM PRESSURE													
150	21.9	680	2270	123.											D
150	22.5	0			2335		2300	55	2.42	2.86	134		120.5	11.	8.9	
150	22.4	680				2230	2250	50	2.17	3.64		114.	123.	11.5	11.3	
		FULL LOAD TO NO LOAD NONCONDENSING 120% STEAM PRESSURE													
180	2.4	680	2230	127											E
180	2.5	0			2290		2275	45	2.1	2.69	136.		125	9.0	7.1	
180	1.5	680				2210	2230	45	2.35	2.86		118	128	7.0	5.6	
		FULL LOAD TO NO LOAD CONDENSING 120% STEAM PRESSURE													
176	23.	680	2270	127											F
180	22.6	0			2365		2340	70	3.06	4.18	136		134	9.	7.1	
180	23.	680				2250	2290	60	2.56	3.84		118	127	16.	12.0	
		FULL TO 20% OF FULL LOAD CONDENSING 100% STEAM PRESSURE													
150	22.2	680	2270	126											G, GRADUAL
150	21.75	136			2310		2300	30	1.32	1.16	132		122.5	9.8	7.1	
150	22.7	680				2260	2270	30	1.3	1.52		123.5	123.5	9.5	3.53	
150	22.6	680	2273	126											G2 ONE STEP
150	22.5	136			2310		2300	35	1.54	1.54	131.5		127.	5.5	4.35	
150	22.8	680				2260	2274	30	1.3	1.74		122.5	125.5	4.5	3.55	

Plate IX.

Turbine #671
Gen. #271361

HARTFORD, CONN. JULY 11, 1911.

COMPOUNDING TEST.

ARMATURE VOLTS	ARMATURE AMPERES	R.P.M.	STEAM	VACUUM			
125.5	680	2240	150	22.75			
122.	0	2290	"	23.6			
125.-	680	2240	"	22.7			
125.8	600	2255	"	22.8	MAXIMUM VOLTAGE - 125.8		
125.2	500	2250	"	22.1	MINIMUM VOLTAGE - 122.		
124.	400	2255	"	21.6	DIFFERENCE - 3.8		
125.5	300	2255	"	22.2			
123.5	180	2255	"	22.1			
125.3	100	2270	"	22.2			
123.	0	2290	"	22.4			
122.	100	2285	"	22.7			
125.7	210	2260	"	23.6			
124.	300	2245	"	23.75			
125.	400	2255	"	23.6			
125.6	500	2250	"	23.2			
123.8	600	2240	"	23.1			
123.3	680	2240	"	22.9			

Plate X.

Turbine # 671
Gen. # 27/961

Hartford, Conn, July 14, 11

4 Hour - Full Load Heat Run.

Time	Terminal		Steam	Vacuum	R.P.M.	Shunt Field			Temperature		
	Volts	Amp				Volts	Amp	Res.	Air	Rheo.	Sh. Field
9.10	125.	680.	121	23.4	2240	65.	2.04	31.86	22.5	24.	33.
.25	126.	680.	141	23.2	2240	67.5	2.08	32.42	23.	28.	35.
.40	125.	680.	150	24.1	2245	68.5	2.08	32.90	24.	30.	36.5
.55	122.	690.	142	23.4	2250	67.	2.02	33.18	24.5	32.5	37.
10.10	1252	680.	150	23.4	2245	70.	2.10	33.3	24.	34.	39.
.25	125.	680.	150	23.3	2250	72.	2.13	33.8	24.	35.5	39.
.40	125.	680.	142	23.5	2180	70.	2.05	34.8	24.3	34.8	39.
.55	125.	680.	150	23.6	2240	72.	2.09	34.4	25.	37.	41.
11.10	1257	680.	150	23.9	2250	72.	2.10	34.3	24.6	37.	41.
.25	125.	680.	150	23.7	2240	72.	2.08	34.6	25.5	37.5	41.
.40	125.	680.	150	23.7	2250	72.	2.07	34.8	25.	37.5	41.
.55	125.	680.	150	23.8	2255	72.	2.075	34.7	25.	38.	42.
12.10	125.	680.	150	23.6	2260	74.	2.13	34.7	27.	38.5	41.
.25	126.	680.	150	23.3	2255	72.	2.07	34.75	27.	39.5	42.
.40	125.	680.	150	23.2	2255	72.	2.07	34.75	26.	39.5	42.
.55	125.	680.	150	23.1	2260	72.	2.07	34.75	27.	39.	42.
1.10	125.	680.	150	22.9	2255	71.6	2.04	35.	26.5	38.5	41.
							Commutation O.K.				
2 Hour Run at 33⅓% Overload.											
1.50	125.	910.	150	22.	2240	79.5	2.15	34.2	26.	32.5	40.
2.05	125.	910.	150	20.8	2245	73.	2.12	34.5	25.5	35.	44.
2.20	125.	910.	148	20.6	2255	72.	2.07	34.8	26.	36.	44.
2.35	125.	910.	127	20.9	2215	73.	2.08	35.1	26.5	37.	46.
2.50	126.	920.	134.	21.5	2210	74.	2.10	35.2	26.	37.	46.
3.05	125.	910.	150	22.	2240	74.	2.09	35.4	27.	37.	46.
3.20	126.	910.	157	22.	2240	75.	2.10	35.7	27.	38.	49.
3.35	126.	910.	150	21.3	2235	75	2.10	35.7	26.5	38.	49.5
3.50	126.	910	150	20.7	2240	74.	2.06	35.9	25.5	38.	49.
							Commutation O.K.				

Plate XI.

Turbine #671
Gen #27361

Hartford, Conn. July 14, 1911.

One hour full load water rate (taken during 4-hour heat run)

Time	Terminal		Steam	Vacuum	R.P.M.	K. W.	Temp. F.A.L. Deg. F.	Water Reading Net	Quality	Water rate actual per K.H. hour	Water rate corr. to dry steam 27" vac.
	Volts	Amp.									
10.52	125	680	150	23.7	2265	85	281.				
10.58	125.	680.	150	23.8	2260	85	279.	306.			
11.04	125.	680.	148	23.8	2245	85	287.	307.			
11.10	126.	680.	150	23.9	2250	85	292.	316.			
11.16	125	680	145	23.9	2250	85	286	306.			
11.22	125	680.	150	23.9	2260	85	266.	303			
11.28	125	680.	150	23.8	2265	85	276	311			
11.34	125.	680.	148	23.8	2265	85	285.	309.			
11.40	125.	680.	150	23.7	2250	85	279.	313.			
11.46	125	680	150	23.8	2250	85	295	304.			
11.52	125	680.	150	23.8	2240	85	289.	313.			
Avg.	125.	680.	149	23.8	2256	85	283.	308 lb. per Hour	.985 sat.	36.33	31.49

Plate XII.

Turbine # 671
Gen. # 27961

Hartford, Conn. July 14, 1911.

Temperatures by Thermometer

	Cold	End of Full Load Run		End of 33⅓% Overload Run		
		Temp	Rise	Temp	Rise	
Air at Machine.	22.5	26.5	4.	26.5	3.	
Armature Surface.	24.5	50.	21.5	60.	33.5	
Armature Core.	24.	48.	20	57.	30	
Commutator Bars	23.5	43	15.5	49.5	23.	
Pole Tip Leading	24.5	50	21.5	60.	33.5	
Series Field	24.	44.	16	53.	26.	
Shunt Field	24.	44.	16.	53.	26.	
Series Shunt	24.	52.	24.	64.	37.	
Armature Coils	24.	42.5	14.5	51.	24.	
Field Rheostat	22.	39.	13.	38.	13.	
Inboard Bearing	23	44.	15.	57	31.	
Outboard Bearing	23.	44	15	57	31	
Interpole	24.	47	19.0	60.	33.	

Full load at various voltages.

Volts	Amp	R.P.M.	
100	850	2240	
110	773	2240	Emergency trip operates at 2480
120	707	2240	9.9% overspeed.
130	654	2240	
140	606	2240	
150	568	2240	
160	530	2240	
175	485	2240	
Operation tried at these voltages and found to be O.K.			

Plate XIII.

TESTS OF 85 K.W. TURBO-GENERATING SETS FOR U.S.S. INGALLS.

SUMMARY.

	GUARANTEED	DATE OF TEST.	STEAM PRESSURE.	QUALITY	VACUUM - INCHES	SPEED - R. P.M.	FULL LOAD WATER-RATE POUNDS PER K.W. HOUR ACTUAL.	FULL LOAD WATER-RATE POUNDS PER K.W. HOUR CORR. TO 27" VACUUM.	HALF LOAD WATER-RATE POUNDS PER K.W. HOUR ACTUAL.	HALF LOAD WATER-RATE POUNDS PER K.W. HOUR CORR. TO 27" VACUUM.	MAX. JUMP % SPEED LOAD THROWN OFF NORMAL CONDITIONS.	STILLED SPEED VARIATION IN % NORMAL CONDITIONS.	MAX. JUMP VOLTS. LOAD THROWN OFF NORMAL CONDITIONS.	VOLTS VARIATION COMPOUNDING TEST.	COMMUTATION - FULL LOAD	DIAG. C - MAX. TEMP RISE FULL LOAD.	EMERGENCY SPEED TRIP IN % ABOVE NORMAL SPEED.
	GUARANTEED		150	1.00	27	2300		36.			3.5	4.5	18.75	8.5		40	10.0
MACHINE #70		4/28-29/11	150	1.00	27.9	2290	36.25	32.6	42.6	38.		1.77	9.2	3.3	1/4 +	22.5 ■	10.7
MACHINE #71		3/11-14/11	150	.985	23.8	2256	36.33	31.49				2.66	14.5	3.8	1/4 *	24. ■	9.9

+ COMMUTATION 1/4 ON OVERLOAD
# " 1/2 " " "
* MAX. RISE 38° " "
■ " " 37° " "

Plate XIV.

NOTES.

The Diesel engine for marine use is attracting world-wide attention. The following paper presents the subject so fully, including the progress made in the development of this engine, that we present the paper in its entirety. We regret that this paper was given to the world too late to be included in our May issue.—EDITOR.

DIESEL ENGINES FOR SEA-GOING VESSELS.

By J. T. MILTON.

Read at the Fifty-Second Session of the Institution of Naval Architects, April 6, 1911.

The possibility of employing internal-combustion engines for the propulsion of large vessels has recently been prominently before those interested in shipping, and as it had been determined to fit several vessels intended to be classed in Lloyd's Register with these engines, the committee of that society determined to obtain as much information as possible in regard to their construction and their suitability for the purpose. As the Diesel engine in some of its forms appeared to be that which was making most progress, they asked for and obtained the co-operation of those firms on the Continent who had given special attention to this form of engine for marine work, and visits were made by the chief engineer surveyor and other officers of the society to each of the firms in question, and much useful information was obtained. The firms were Messrs. Burmeister and Wain, of Copenhagen; the Nederlandsche Fabriek van Werktuig en Spoorweg Materieel, of Amsterdam; Messrs. Carels Freres, of Ghent; the Maschinenfabrik Augsburg Nurnberg, of Nurnberg;

Messrs. Sulzer Brothers, of Winterthur; Société John Cockerill, of Seraing; and Messrs. Schneider and Cie., of Le Creusot, each of which, besides having considerable experience in the design and manufacture of internal-combustion engines, had either completed or had in course of construction some Diesel engines for marine purposes. The British Diesel Engine Company were consulted, and a visit was also paid to Dr. Diesel, with the same object in view.

This gentleman and the principals of the above-mentioned firms courteously and freely gave full information upon every point upon which their previous work had given them experience.

British Firms and the Diesel Engine.—Since then several British firms have decided to make Diesel marine engines, and as numerous inquiries are being constantly made by those interested in the subject, the present paper has been prepared. It is hoped that the information contained in it and that brought out in the discussion will be of service to shipowners, engineers and shipbuilders, and will not be without interest to underwriters also.

It need hardly be pointed out that the inducement to forsake the tried and trustworthy servant, the steam engine, for ordinary sea-going vessels will be mainly, if not solely, the question of fuel economy; but this point, important though it is, would not of itself warrant a change to a new type of engine unless equal certainty of continuous efficiency on the voyages to be undertaken was provided, that is, as little risk of accident to machinery and as great facility for using temporary expedients for reaching port in case of breakdown of part of the machinery. There must also be a prospect of a reasonable cost of upkeep. To each of these points some attention will be given in this paper.

It is evident that, apart from the question of the relative cost and facilities for obtaining supplies of oil fuel and coal, an internal-combustion engine using oil possesses many advantages for marine work over one using gas made from coal.

Even for land installations the same thing may be said, inasmuch as with oil the comparatively bulky gas producers and scrubbers are not required. There is no trouble with dust in the cylinders, and there should be none with tar on valve faces, etc. Even as fuel for marine steam boilers the advantages of oil over coal are very great. Weight for weight it has at least 50 per cent. higher evaporative efficiency. It occupies less space per ton than coal. It can be carried in double bottoms and other spaces unsuitable for coal or cargo. It requires no trimming, and lessens the numbers of stokers necessary. Owing to the higher temperature produced and more perfect combustion obtainable, more power can be realized by its use from the same size of boiler. There is greater regularity of power than with coal owing to there being no fire cleaning necessary. It can be quickly pumped from the stores into the bunkers, and the labor and dirt of coaling ship are avoided. Yet with all these advantages oil fuel for ships' boilers is very little used as compared with coal. Leaving war vessels out of account, oil fuel is only used on shipboard in those places where the natural advantages render its use more economical than coal, and by vessels which trade regularly to ports where supplies can be obtained. For the ordinary cargo steamer which has to seek employment all over the globe, coal is still the necessary fuel.

It is said that unlimited quantities of oil are known to exist well distributed throughout the tropical and temperate regions of the world, and that, given a regular and steady demand for it, commercial enterpri e will place ample supplies upon the market. Undoubtedly, for many years most of the oil raised was for the supply of the refined products, petrol, lamp oil, and to a less extent lubricating oil, the residuum after these had been distilled from the crude oil being a by-product which had either to be thrown away or used as fuel. In these conditions, with a comparatively limited supply of residuum, its price would naturally be determined, not by the cost of raising oil, but by the value of the coal with which it had to compete. The market for refined oils, although large, is not

unlimited, and it is thought by some that they are at present over-produced. The cost of distillation is a very considerable factor. Considerations of safety render it necessary to use in ships' boiler furnaces only fuel oils having a comparatively high flash point. This practically shuts out from marine-boiler use those crude oils which do not contain sufficient of the lighter hydrocarbons to make it profitable to distil them, but which contain sufficient to give them a comparatively low flash point. If the oils are to be used in closed cylinders instead of in ordinary boiler furnaces there is not the same necessity for demanding a high flash point. Experience has shown that even the extremely volatile "petrol" can be safely carried in bulk on board ship.

From the above it may be considered that if internal-combustion engines of the oil-burning type become common in sea-going vessels, crude oil, rather than the residues from distillation, will form the major portion of the fuel employed, raised, perhaps, in places where it may be profitable to distil some portion, but it will also be profitable to employ crude oils that are not suitable for distilling.

Oils for Use in the Diesel Engine.—In regard to this matter it may be well to state here that it is claimed for the Deisel engine that any oil can be used in it which will burn completely, leaving no residue. If this is so, the different qualities of oil must require special experience to know how to adjust the fuel valve to spray them properly, so as to obtain a suitable combustion that will on the one hand consume them sufficiently slowly to prevent an undue rise of pressure, such as occurs in gas and petrol engines, and on the other will permit of their total combustion in the engine. There have, however, been objections taken to this view. Mr. Paul Rieppel, of the Maschinenfabrik Augsburg Nurnberg, of Nurnberg, who has had much experience with internal-combustion engines, has made some elaborate investigations on this point, and, after experiments with many kinds of oils, he has come to the following conclusions:

The various hydrocarbon oils which occur either as natural

products or as the result of distillation, either of brown coal, ordinary coal, or crude oils, may be divided into two classes, which differ very much in their behavior when raised to a high temperature. Those of one class, when raised to such a temperature as is obtained by the compression in Diesel engines, readily decompose into free hydrogen and heavier hydrocarbons, those of the other class at first only vaporize, or partially vaporize, and require much higher temperature than that due to the compression for their decomposition. In those of the first class the hydrogen, because of its easy ignition, burns first, and the resulting heat enables the remainder of the hydrocarbons to become completely burnt, the total combustion taking an appreciable, though small, amount of time. In the second, a very high temperature is required to initiate combustion, which, when commenced, approximates to an explosion, owing to part of the oil having been made gaseous becoming mixed with the compressed air, and so constituting an explosive mixture. The latter oils, therefore may be either not properly consumed or they may be burned with explosive violence.

It should be mentioned that Mr. Rieppel states that even these oils can be used in the Diesel engine by means of special alterations in the design, but it would appear that they could not be used in an engine adjusted for the more suitable oils. In general the oils which are not suitable appear to be the result of distillation of ordinary coal, so that this matter is of more importance to the users of land engines, whilst the oils made from brown coal or from natural crude oils appear to be well suited for Diesel engines.

There is, however, another point which requires attention. Some of the crude oils, and therefore all the residues from their distillation, contain combined sulphur. Some oils are credited with containing as much as 2 per cent. This element, especially at high temperatures, has an affinity for copper, and it has been found that where oil containing sulphur has been used the valves and fittings coming into contact with the oil must not be made of copper or copper alloys. Fortunately

cast iron appears to resist the action of the sulphur, and valves, etc., made of this material have stood well where brass valves have failed. This point shows also the advisability of making all fuel pipes of steel or iron, instead of using copper, unless it is certain that the fuel supply will not contain sulphur.

Whilst dealing with the question of the quality of oil required, it may be advisable to state that extended experience has shown that the requirements of Lloyd's Rules for oil-fuel bunkers, etc., have proved their sufficiency for safely dealing with heavy oils suitable for boiler fuel, having a flash point of not less than 150 degrees F. Such oils have been through refineries, and all light hydrocarbons have been distilled from them. Any gas arising from them at even tropical temperature must be very small in quantity and entirely negligible, provision is made for dealing with any possible leakage from any of the oil-fuel compartments, and fore and aft peak tanks and double bottoms under cargo holds may be safely used for the fuel. If, however, oils are to be used for engine fuel which have a low flash point, or which give off a perceptible amount of gas, it will be undesirable to carry them in double bottoms under cargo holds or in spaces from which a leakage would permit of vapors or gas penetrating into cargo holds, or other closed spaces where they would be a source of danger.

It must be remembered in this connection that some of the hydrocarbons, cresote, for instance, have a most objectionable and penetrating odor, capable of contaminating and spoiling many kinds of fine cargoes. However, as "petrol" can be safely carried in bulk, suitable provision can be made for any kind of oil to be carried without risk either to the vessel or her cargo.

Oil Consumption in the Diesel Engine.—It may be well to state here what is claimed for the Diesel engine in the way of consumption. In an ordinary steam engine the power is generally reckoned as indicated horsepower. This is the work performed by the steam on the piston, and is the gross power obtained. It has to overcome the friction of the mech-

anism, work the slide valves and the pumps, and only about 85 per cent. in round numbers is transmitted to the shaft.

In the Diesel engine the indicated horsepower has similarly to overcome the friction of the mechanism; it has to work the fuel pump, the mechanism for actuating the valves, and to supply the compressed air necessary for injecting the fuel. In the two-stroke cycle also it has to work the scavenging pump. These take up more of the gross power than the accessories do in a steam engine, and hence a less proportion of the gross, or indicated power, is transmitted to the shaft than in a steam engine. For this reason the power of a Diesel engine is more usually expressed as its brake horsepower, that is, the power usefully exerted outside itself.

It is usually claimed that the oil consumption per brake horsepower per hour is 0.4 pound when the engine is working at full power, and when working at somewhat lower powers the rate of consumption is not much increased.

If one assumes that in a modern steam engine the consumption of coal is 1.25 pounds per I.H.P. per hour this corresponds to about 1.47 pounds per B.H.P., so that the weight of fuel to be carried for the same voyage in a vessel fitted with Diesel engines would be only 28 per cent. of that of the coal necessary with ordinary steam engines.

Types of Engine Available.—We will now turn to the engine itself. Its principle of working is generally known. It is made in three forms for marine purposes, namely, as a four-stroke cycle single-acting engine, a two-stroke cycle single-acting engine, and a two-stroke cycle double-acting engine. An essential feature of these engines is that they require, besides their own cylinders and pistons, an auxiliary air compressor capable of producing a pressure of about 700 pounds per square inch.

The Four-Stroke Cycle Engine and its Action.—In the four-stroke cycle engine the cylinder cover contains a fuel valve, a compressed-air-admission valve, one or more ordinary air-admission valves, and one or more exhaust valves. All these valves are actuated, that is, opened by means of cams

fixed to a cam shaft, and are kept closed by powerful springs when the cams are out of action. The cam shaft is driven by a two-to-one gear, that is, it makes only one revolution for two revolutions of the engine crank shaft. Broadly speaking, the cams are so arranged that the air-admission valves are open during one whole down stroke, and the exhaust valve during one whole up stroke, but actually a little "lead" is necessary. The cams for the fuel valve and the compressed-air valve are so arranged that only one of these can be in operation at a time, so that when either is in use the other is entirely inoperative. In ordinary running the fuel valve is opened at the proper time when the piston is at the top of its travel, and is closed again when about one-tenth of the downward stroke has been made. The compressed-air valve is only used for starting purposes, and it is kept open for a longer period, say for half or even more of the stroke, its range of opening being made to depend upon the number of cylinders used, so that when these valves are in gear there is no position of the engine in which there is not at least one of them open.

In starting the engine these valves are put into gear, and the fuel valves are consequently put out of action. When the engine has made one or more complete cycles, the compressed-air valves are put out of gear, the fuel valves commence their work, and the engine then continues its motion, working under "fuel" conditions. As the air-admission valve gear is in full action during the starting operations, the full compression would have to be overcome in each cylinder in turn if it were not for a special arrangement made to relieve part of the pressure in order to facilitate starting. This is put out of action when the fuel admission is put into gear. Commencing with a piston at the top of the cylinder, the four-stroke cycle is as follows:

First Downstroke.—The ordinary air-admission valve is open during the whole stroke, and the cylinder becomes filled with atmospheric air at the ordinary atmospheric pressure.

Second Stroke.—The air valve is closed, and the piston returns to the top of the cylinder, compressing the air which had been drawn in during the previous stroke. The clearance is so proportioned that in ordinary working at full speed the pressure becomes about 500 pounds per square inch, and the temperature is at the same time very much raised. The compression is not quite adiabatic, as the cold cylinder walls must abstract a little of the heat from the air. If it were truly adiabatic the temperature of the air would be raised from, say 60 degrees F. to 1,000 degrees F.

During this stroke a quantity of fuel has been pumped by the fuel pump into an annular space round the fuel valve. When the piston is at the top of the stroke the fuel valve is raised, and at the same time cold air from the air-compressor reservoir at a pressure of 700 pounds per square inch blows the fuel into the cylinder, which contains hot air at a pressure of 500 pounds per square inch. The construction of the fuel valve is such that the oil is "pulverized" or "atomized," that is, it is divided up into a spray of very fine particles. These, upon coming into the very hot air in the cylinder, ignite, and the heat produced by the combustion increases the volume or the pressure of the air. When the adjustment of the valve is correct, the admission of the fuel and the combustion proceed at such a rate that they are almost completed during the time taken for the piston to travel one-tenth of its stroke, and during this period the pressure of 500 pounds per square inch is maintained.

Third Stroke.—The third stroke of the cycle commences with the combustion of the fuel as mentioned above, after which, during the remainder of the stroke, the hot gases in the cylinder expand until the end of the stroke is reached.

Fourth Stroke.—The return of the piston constitutes the fourth stroke, and during this time the exhaust valves are open and the burnt gases are expelled from the cylinder. After this the cycle commences afresh.

In the two-stroke cycle single-acting engine the cylinder covers are similarly fitted with fuel valves and compressed-

air valves for starting purposes, but the ordinary air-inlet valves and exhaust valves are replaced by scavenge air valves. All these valves are actuated by cams; the cam shaft, however, in these engines rotates at the same speed as the main engine shaft.

The pistons are made somewhat deeper than the total length of stroke. At the lower end of the part of the cylinder barrel uncovered by the movement of the piston, there are numerous ports leading into the exhaust passage. These ports have a vertical dimension of about one-seventh of the stroke.

The Two-Stroke Cycle.—The two-stroke cycle is as follows:

First Stroke.—When the piston is at the bottom of the stroke the cylinder is full of pure air at atmospheric pressure, which air has just been admitted through the scavenge valve. During the up stroke the air is compressed up to 500 pounds pressure per square inch, precisely as in the compression stroke of the four-stroke cycle engine.

Second Stroke.—The second stroke commences at the top center by the admission of fuel sprayed in by compressed air precisely as in the previously-described engine, and the stroke proceeds in exactly the same way until the piston has traveled about six-sevenths of the stroke. At this point it commences to uncover the exhaust ports through the cylinder sides. So much of the hot gases escape through these ports as to reduce the pressure in the cylinder to about that of the atmosphere. The scavenge valves are then opend and fresh air under pressure is admitted into the cylinder, blowing out the remainder of the burnt gases into the exhaust passages. With these gases some of the scavenge air also passes into the exhaust. By the time the piston has reached the bottom of its stroke, the scavenge valves are closed, but the cylinder is left full of clean air ready for the compression stroke to commence.

Arrangements for Supplying the Scavenge Air.—There are different arrangements made by different makers for supplying the scavenge air. In some designs each main cylinder has its own air-compressing arrangement and receiver. In

other designs sometimes one and sometimes two air pumps are provided, sometimes worked by cranks from the crank shaft, and in some cases by levers similar to the method of working air pumps in ordinary marine steam engines.

There are also different methods of applying the scavenge air, one allowing it to enter through special valves in the cylinder cover, another by admitting it at one side of the bottom of the cylinder through ports uncovered by the piston in the same way as the exhaust ports are uncovered. In this latter case the scavenge ports are on one side and the exhaust ports on the other.

The volume of the scavenge-air pumps is considerably greater than that of the cylinders, the proportion being in general not less than 1.8. This is necessary to ensure that all the burnt gases are swept out of the cylinder. As the full quantity of air dealt with by the pump must pass from the reservoir into the cylinders every revolution, the pressure to which the air in the reservoir attains depends upon the scavenge-valve openings. When these are large a less pressure is required to force the air through them than when they are small. Hence the larger the openings of these valves the less load there is thrown on the scavenge-air pump.

In some designs the scavenge-air pressure is as much as 7 to 8 pounds per square inch, in others it is as low as 3 to 4 pounds above the atmosphere.

Pistons.—In the preceding engines the pistons are of the trunk form. In the double-acting two-stroke cycle engine they are necessarily made of box form, and are fitted with piston rods, which, as they pass through the burning gases at the lower side of piston, have to be specially cooled. The pistons also, in general, have to be cooled by either oil or water circulation.

Water is the best cooling medium, as its specific heat is about three or four times that of oil, but some prefer oil, as any leakage from the water circulation washes off the lubrication of any of the rubbing parts which it touches.

The admission, fuel valves, etc., are designed for both the

top and bottom of the cylinder. The exhaust ports are in the middle of the length of the clyinders, and the pistons, as in the single-acting engines, uncover the ports at nearly the end of the stroke.

In small engines of the single-acting type the pistons are not water or oil cooled, as it is found they may be kept at a sufficiently low temperature by their contact with the cylinder sides which are water cooled. A heated piston is not so objectionable as it is in ordinary gas or oil engine with timed ignition, as in the Diesel engine preignition cannot occur. The main objections are that, with a large piston overheating of the crown may be the cause of structural weakness, and that the expansion of the metal by its high temperature renders it necessary to make the piston crown initially smaller than the cylinder bore.

This has to be arranged in all engines, and the exact amount of allowance is one of those points in which experience is the only guide. It may be said here that this is a matter of extreme importance in those engines in which there are no piston rods. This will be again referred to further on.

Experiments are being made in the case of an engine with a large diameter of cylinder to ascertain whether it is practicable to run it without special piston cooling.

Air Compression.—In all the types of engines highly compressed air is needed for starting purposes and also for the fuel injection. This has to be supplied by an air-compressing plant worked by the main engine. The compression is sometimes performed in two stages, although a three-stage arrangement is generally used. The compressed air is cooled at each stage. The volume of the compressor is such as to provide a small surplus each revolution over that required for the fuel admission in continuous working. This surplus is stored in a reservoir constructed usually as a battery of seamless-steel bottles. These are tested by hydraulic pressure to 120 atmospheres. A safety valve is provided loaded to 60 atmospheres. The compression of the air is attended by

the deposition of moisture from it, so that means of draining the steel bottles should be provided.

Choice of Engine.—Naturally there are advantages and disadvantages with each type of engine, and a judicious consideration of these should determine which is the more suitable type to use in any particlar case. The two-stroke double-acting engine will be higher than a single-acting one with the same diameters of cylinders and stroke, but the power will be obtained with a less number of cylinders. On the other hand, there is considerably more complexity in the valve arrangements and a probability of difficulty with piston-rod stuffing boxes, to say nothing of the trouble of cooling the pistons and rods by water or oil circulation. There is also likely to be considerable difficulty owing to want of access for overhauling. It should be stated that no experience has been had, as yet, with large engines of this type.

A Comparison.—Comparing single-acting engines of the two-stroke and four-stroke types, the former require only half the number of cylinders which are requisite in the latter, either to produce the same power or the same degree of uniformity of turning moment. The four-stroke, therefore, means a longer engine and necessarily a heavier one also. The valve gear of the two-stroke engine being actuated by a shaft with the same rotational speed as the crank shaft is simpler than that in a four-stroke engine, and the reversing arrangements are much less complicated. The two-stroke, however, requires the addition of the scavenge arrangements which are absent from the four-stroke, and the necessity for supplying the energy for working these makes the mechanical efficiency less. On the question of efficiency, however, it may be urged that the four-stroke engine has to overcome the friction of the piston, etc., for what may be called two idle strokes out of every four, and this must, to some extent, counterbalance the energy necessary to work the scavenging pumps. In the four-stroke engine all the hot used gases have to escape past the exhaust valves, which thus may become abnormally heated. On the other hand, in the two-stroke engine,

they have to pass the bars between the exhaust ports, and it is thought by some that, although these parts of the cylinder are water jacketed, they must become overheated and lose their accuracy of surface, and it must be remembered that all the piston rings have to pass these bars every stroke. Extended experience will be required to settle all these points. It may be mentioned that an engine is being made on the four-stroke system in which the major portion of the exhaust passes out of the cylinder through ports precisely as in the case of the two-stroke engine, leaving only a part of the burnt gases to be pushed out of the cylinder through the ordinary exhaust valves.

Indicator Diagrams Compared.—It is instructive to carefully examine the indicator diagram of a Diesel engine and to compare its compression and expansion lines with adiabatic curves.

When it is remembered that the circulating water abstracts about one-fourth of the total heat of combustion of the fuel it will be seen that adiabatic conditions must be widely departed from.

At the commencement of the compression the air temperature will not be very different from that of the atmosphere, and the air will no doubt receive some heat from the cylinder walls. During the compression, after its temperature rises above that of the cylinder walls, some heat must be given up so that the pressure will not quite reach that due to adiabatic compression.

During expansion, when its temperature is so very high, it must impart a very considerable amount of heat to the cylinder walls, so that after combustion is complete its temperature, and therefore its pressure, must fall below that due to adiabatic expansion. The fact that during nearly all the expansion the pressure is above that of the adiabatic curve shows that combustion must be taking place almost throughout the stroke, and to such an extent as to supply more heat to the gases than is being abstracted by the cylinder sides. It is necessary for complete combustion to take place in order to prevent the fouling of the cylinders by the accumulation

of carbonaceous or tarry residues, and it is this necessity for completion of the combustion, which may be termed slow compared with the more rapid burning of the major part of the fuel admitted, which limits the amount of fuel usable per stroke. It can readily be understood that the physical condition of the pulverization of the fuel is of the greatest importance. If the spraying is not fine enough, the combustion will not be sufficiently rapid, whilst if it is too fine it may be so fast as to raise the pressure above the 500 pounds per square inch, which should be the maximum obtained. It would thus appear that different adjustments of the fuel valve will be required with different qualities of oil, which may vary in their chemical composition and also in their viscosity. Further, the conditions also will to some extent depend upon the speed of the engines.

In small cylinders only one fuel valve is employed. It is placed centrally. In very large cylinders it is thought that more than one must be used in order to provide for a more uniform distribution of the fuel into the compressed air, but the conditions determining the number actually necessary are matters which can only be ascertained by experience.

From the results obtained with a Carels Diesel engine in some tests made by Professor H. Ade Clark at Ghent, which were published in "Engineering," August 7th, 1903, it appears that the total amount of air used per stroke was 3.31 times that theoretically necessary for the complete combustion of the fuel. This shows that the limiting quantity of fuel which can be consumed in the engine depends not so much upon the oxygen supply as upon the time element, and, as above indicated, must be considerably influenced by the physical conditions of the spraying.

The large amount of air used must have a considerable influence in lessening the temperatures obtained by the combustion, but these would be almost the same whether the dilution was with clean air or whether some small part of it was composed of already burned gases. With an admixture of burned gases, such as is always present, for instance, in an

unscavenged gas or petrol engine, the combustion would probably be somewhat slower; but it is worth investigation whether under certain conditions, which of themselves necessitate slower running of an engine, complete combustion, and the consequent prevention of fouling of the cylinders, would not be obtained in cases where a less than normal oxygen supply was obtainable. This point will be again referred to later on. It may be noticed that in a four-stroke cycle engine there is always a small proportion of burned air present in the charge, as the clearance space must be left full at the end of the exhaust stroke.

Determining the Number of Cylinders.—In dealing with large powers it is not to be wondered at that there are different ideas amongst engineers who have had experience with Diesel engines as to the best course to adopt—namely, whether to use a large number of small cylinders or a less number of larger ones. For land engines, where very heavy flywheels can be adopted, the conditions are altogether different from those obtaining on board ship, and one, two, three, or any number of cylinders can be employed, according to the choice of the builders, provided the flywheels are made sufficiently heavy to give the steadiness of motion requisite. On board ship, however, flywheels are not desirable, and very heavy ones are altogether out of the question owing to the slowness of maneuvering they involve.

In determining the number of cylinders requisite in any given case, it will be well to examine the torsion-moment diagrams.

It will be seen from the indicator diagrams that when working at full power the forward pressure starts at 500 pounds per square inch, and at the end of the expansion it falls to less than one-tenth of this amount; the back pressure similarly varies from atmospheric pressure at the commencement of the return stroke to about 500 pounds at its termination. These conditions are altogether different from anything occurring in a steam engine. The effect of the inertia of the reciprocating parts, however, is considerable at full speed,

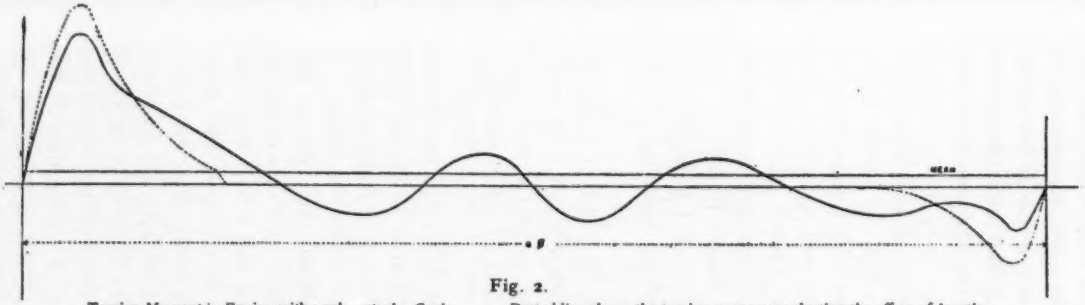


Fig. 2.

Torsion Moment in Engine with 1 cyl. 4 stroke Cycle. Dotted line shows the torsion moment neglecting the effect of inertia.

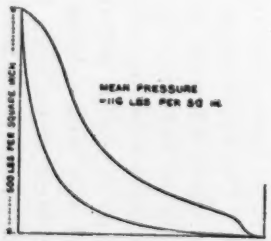


Fig. 1.
Indicator Diagram of Diesel 2 stroke Cycle Engine.

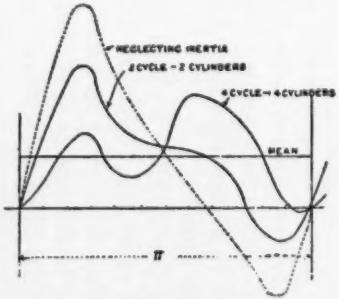


Fig. 3
Combined Torsion Moment in 4 cyl. 4 stroke Cycle, and in 2 cyl. 2 stroke Cycle Engine. Dotted line shows the combined moment in either case if inertia is neglected.

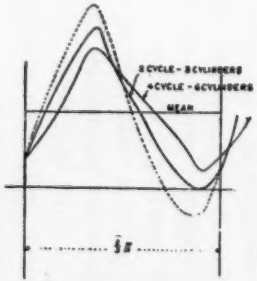


Fig. 4
Combined Torsion Moment in 6 cyl. 4 stroke Cycle Engine, and in 3 cyl. 2 stroke Cycle Engine.
Dotted line shows the combined moment in either case if inertia is neglected.

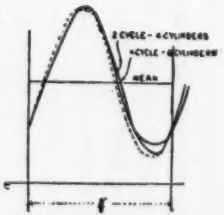


Fig. 5
Combined Torsion Moment in 8 cyl. 4 stroke Cycle, and in 4 cyl. 2 stroke Cycle Engine.
Dotted line shows the combined moment in either case if inertia is neglected.

lessening the maximum load upon the crank pin at the commencement of the forward stroke and augmenting it towards the end, so as to tend to give a more uniform distribution of load throughout the stroke, and similarly during the compression stroke the inertia effect also tends to uniformity.

As, however, in starting the engine the motion has necessarily to commence at a low speed at which the inertia effect is negligible, it is desirable to examine the torsion-moment diagrams with the inertia neglected as well as when under full-power working conditions. The figures showing the moments under normal working conditions are reproduced from some kindly supplied by Messrs. Sulzer Brothers. Upon some of them have been drawn in dotted lines the conditions which apply upon starting without allowing for the inertia.

In the four-cycle engine it will be seen that with one cylinder only the torsion moment varies from thirteen times the mean effective moment in a positive direction to nearly four times the mean in a negative, a total fluctuation of about seventeen times the mean.

When four cylinders are used, Fig. 3 shows that the torsion moment varies from about two and a quarter times the mean moment to one-tenth of the mean moment in a negative direction, the total range of stress being 2.3 times the mean. In the corresponding two-cylinder two-stroke cycle engine the range is greater, being from a maximum of 2.8 to a minimum of —.68 time, or, approximately, three and a half times the mean stress. The figure shows that in this case the four-stroke cycle engine will work more smoothly than the other.

Considering the four-stroke cycle engine with six cylinders, which is the smallest number of cylinders likely to be proposed with this type of engine, and the corresponding two-stroke cycle engine with three cylinders, Fig. 4 shows that in the former the stress varies from 1.85 to .25, a range of 1.6 times the mean, and in the latter from 2.1 to zero. The advantage again is in favor of the former.

Comparing a four-stroke engine with eight cylinders with a two-stroke single-acting with four cylinders, and a double-acting engine with two cylinders, we see from Fig. 5 that all three engines have about equal variations of torsion moment, the range being from 1.75 to .37 in each case. With these numbers of cylinders the motion approximates in uniformity to that obtained with a steam engine with two cranks.

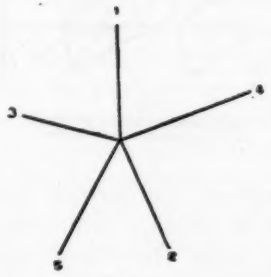


Fig. 6.

Fig. 7.
Torsion Moment on end length of crankshaft of a 5 cyl. Engine.

If we similarly examine the diagrams of a four-stroke cycle engine with twelve cylinders, a two-stroke with six cylinders, and a double-acting with three, we find in all three engines a fairly uniform torsion moment, the ratio of maximum to mean being not greater than 1.15, which is less than is usually obtained in an ordinary triple-expansion steam engine. It may fairly be said that with these numbers of cylinders there is nothing further to be desired regarding steadiness of motion.

In deciding upon the number of cylinders to employ in any case considerations of the turning moments point to a minimum of six cylinders with the four-stroke cycle of three cylinders with the two-stroke cycle, and that the larger number of cylinders employed the smoother and better the engines will work. Against this, however, is the fact that the larger the number of cylinders the greater will be the number of moving parts and the more the attention required. More space will be demanded for the engine, which will also become

more costly. With the larger number of cylinders, however, in general, owing to the more evenly distributed stresses, the smaller will be the size required for the shafting for the same power.

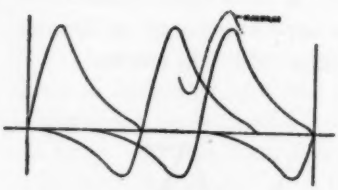


Fig. 8.
Torsion Moment on the 3rd length of crankshaft of a 5 cyl. Engine.

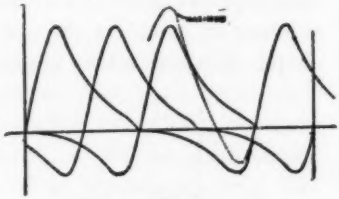


Fig. 9
Torsion Moment on the 4th length of crankshaft of a 5 cyl. Engine.

When very large powers are required, the question will arise as to the upper limit of size of cylinder permissible. On this point there is no actual experience. It is stated that there need be no limits of size other than those demanded for strength considerations. With larger cylinders, however, it is thought that multiple-fuel admission valves will have to be employed. In general it is thought that for very large powers it will be advisable to use more than one line of shafting, keeping the sizes and numbers of cylinders on each shaft within reasonable limits.

The Question of Handiness for Maneuvering.—One point in connection with the number of cylinders is the question of handiness for maneuvering. The engines have to be started with compressed air, and when they have commenced to move, the air arrangement has to be put out of action and the fuel supply put into gear. In the six-cylinder engines of the *Vulcanus* the starting arrangements have been subdivided, so that the change over can be first made with three cylinders, whilst the other three are still working with compressed air, after which the change can be made with the remainder. The experience of the readiness of handling which this method provides is so satisfactory that it will be repeated in some larger engines being made by the same firm.

Another point in connection with the best number of cylinders to employ in any particular case arises when consideration is given to the possibility of working the engines in the event of the breakdown of one of the cylinders, connecting rods, etc.

In a steam engine many casualties may occur in which one cylinder only is disabled, and the engine can still be worked by the remaining two without a serious loss of power or speed, each of the remaining pistons getting a higher pressure upon them than when working under normal conditions. In the Diesel engine, under similar conditions, one cylinder may be disabled, and the others will still be able to work at their full pressure; the loss of work then will be less in proportion to the number of cylinders employed.

Practice Followed in Large Engines Under Construction.—It may be here mentioned that in the four-stroke cycle engines actually made, or in process of building, for large ships either six or eight cylinders are being used per shaft. In the two-stroke cycle single-acting engines four cylinders seem to be preferred, whilst the double-acting engines being built have three cylinders. Both five and six cylinders have been proposed for the two-stroke cycle.

The Working of Auxiliaries.—In a steam vessel, besides the main propelling machinery, there are a number of auxiliaries, all worked by steam. It remains to be considered what would be the best way to provide for doing their work in a vessel fitted with Diesel propelling machinery.

The more important are the following:

(1) Steering gear.
(2) Whistle.
(3) Donkey pumps for bilge and fire service.
(4) Electric-light machinery.
(5) Distillers.
(6) Steam-heating apparatus.
(7) Water-ballast pump.
(8) Winches.
(9) Windlass.
(10) Ventilating apparatus in passenger vessels.

Of these the first six and the last are always required at sea; the remainder are generally only needed when the vessel is either in or near port.

Probably for these latter the best solution is to continue the present practice of working them by steam from a donkey boiler, which may be fired with oil fuel. This boiler can be put into action only when in or about to enter port. For the remainder there are some alternatives.

The electric-light machinery presents no difficulty, as it can be worked by a small oil engine of either Diesel or other type, using the same kind of fuel as the main engine, resulting in a similar economy of fuel as will be obtained with the main engine.

This engine might also be used for working the pumps for bilge and fire purposes, as these pumps are only required on emergencies when part of the electric light may be dispensed with. The ventilating arrangement can be electrically driven, the motive power being the same engine as is used for electric light. The ballast pump may be a steam pump worked from the donkey boiler, but it may be dealt with by working it by means of a small separate oil engine. There remain the steering gear, whistle, distiller, and heating appliances.

The alternatives proposed are:

(1) Having a donkey boiler separately fired by oil fuel continuously at work at sea for all these purposes.

(2) Utilizing the heat of the exhaust gases by passing them through an auxiliary boiler and raising sufficient steam for the purpose.

(3) Working the steering gear and whistle by compressed air from the first stage of the compressor, which would be made larger for this purpose.

The second alternative has much to recommend it in view of the great quantity of heat which passes off with the exhaust gases. According to professor A. Clark's tests previously mentioned, nearly one-third of the total heat of combustion passes off in this way. In the four-stroke cycle engine these gases have a temperature of about 700 degrees F., and are,

therefore, capable of giving a good, useful effect in a boiler. In the two-stroke cycle, however, owing to the dilution of the hot gases with the excess of the scavenging air, the temperature will be much less, and will probably be not greater than 400 degrees F.

Auxiliaries Required by the Engine. Upkeep of the Engines.—Besides these auxiliaries the Diesel engine requires some of its own. The quantity of circulating water required to keep the cylinder cool is very large. This is probably best supplied by a pump worked from the main engines, but in view of a possible derangement of this pump it will be desirable to have also a separate connection with the ballast pump. The air compressors for supplying the compressed air used for injecting the fuel will also be worked by the main engine, but a supplementary compressor will be necessary to keep up the supply of air which is used in maneuvering. This must be worked by a separate engine.

Further, in the event of the receivers losing the air pressure from any cause, it is necessary to have another small engine which may be started without compressed air (say a small paraffin engine), in order to obtain a sufficiency of compressed air, preferably in a small separate reservoir, to start the auxiliary compressor engine.

The size of the auxiliary compressor demands attention. Where an ample volume of reservoir is provided a comparatively small compressor is sufficient, as it can be kept working continuously whilst the demand for air for maneuvering is intermittent. If, however, it is desired to consider it as a provision against a breakdown of the main compressor it must be made nearly as large as the main. The quantity of air delivered through the fuel valve depends upon the time the valve is open; and as the proportion of this to the time taken for a revolution of the engine is fixed, it follows that each cylinder will use about the same quantity of air per minute irrespective of the number of revolutions; that is, the air supply required will be the same whether the engine is running fast or slowly. If, therefore, a small auxiliary com-

pressor is used for the main engines, the only way to keep up the pressure necessary to inject the fuel will be to shut off some of the main cylinders entirely, working only with a similar proportion of the whole to that which the capacity of the auxiliary compressor bears to that of the main compressor.

As a security against breakdown of the main compressor, therefore, it would appear, that it is advisable to have it designed so that all working parts are easily accessible, and to provide a spare piece for each part which is likely to become deranged.

Upkeep of the Engines.—With regard to the upkeep of the engines much can be said. Up till recently these engines have been designed to meet conditions of service on land differing very considerably from those at sea. On land it rarely happens that the engines are worked continuously day after day without stopping, and still more rarely without a weekly stop. These give opportunities for cleaning and for small adjustments. At sea they are required to work continuously. The present marine steam engine is the result of many years' experience and evolution, and if the Diesel engine is to be successful at sea it must be made in accordance with marine-engine practice wherever possible.

On land it is customary to have the connecting rod working on a gudgeon secured inside the trunk piston. The piston itself serves as a guide. Marine engineers are accustomed to have the gudgeon exposed to view; its adjustment can then be quickly and accurately accomplished, and it can be ascertained at every stroke whether its lubrication is satisfactory.

In a double-acting steam engine the alternate thrust and pull of the connecting rod always tend to keep the guide surfaces in contact; that is, the load on the guide is always one way, and if there is any slack there is no blow and noise. In a Diesel single-acting engine the impluse on the down stroke and the compression on the up stroke require the guides to be on opposite sides; hence, if there is the least slack due to wear in the guides, there is a knock both at top and bottom

of the stroke. This, for an engine to run continuously, necessitates adjustable guides. The net result is that the gudgeon should be outside the piston, and should have separate adjustable guides, as in an ordinary steam engine. In other words, a piston should be used as a piston only, and should not fulfil the dual functions of piston and guide.

In land engines it is a satisfactory arrangement to have the guides bored out to cylindrical surfaces, which can be made axially true with the cylinders. This would not be satisfactory for long in a marine engine, as, owing to the wear and possibly to the yielding of the thrust, the crank shaft gradually works forward. If the guide surfaces are plane, a little wear of this kind does not give trouble. Too much attention cannot be given to the lubrication of the pistons in the cylinders. In the steam engine practically none is required, but in the Diesel engine it is necessary that the whole of the rubbing surfaces should be well lubricated. The lubrication is forced, and the arrangement should be such that, for each part requiring it, the oil supply, drop by drop, should be visible so as to render it positively certain that no deficiency is occurring. It is desirable also that the water circulation of each part should be controllable.

In marine engines built crank shafts have almost entirely superseded solid ones, and marine engineers are not likely to go back from their use. This will necessitate the strokes of Diesel engines being made longer, in proportion to the diameters of the cylinders, than is usual in land practice.

Much more consideration will have to be given to the accessibility of every working part than has been done in some designs. Every part of the engine which requires periodical inspection or occasional adjustment must be made easily accessible. Provision must be made for the possibility of renewing the crank shaft without lifting the engines out of the ship.

In short, the Diesel marine engine should be Diesel only as regards the cylinders and their accessories, and should

be of the ordinary marine type as regards all the rest of the engine.

In some designs of two-stroke cycle engine each cylinder has been arranged with its separate scavenge pump. In other designs one large double-acting pump supplies all the cylinders. In the former arrangement if one cylinder fails from any cause the remainder are not interfered with, and if a scavenge-pump valve gives out, only the one cylinder which the pump serves will be disabled. Where, however, one large pump serves all the cylinders, while it is recognized that the arrangement of the pump with one piston and one mechanically-worked valve is very simple and mostly unlikely to get out of order, yet it must be admitted that its failure, if it does occur, will occasion a complete stoppage of the engine.

Possibly in recognition of this, some recent designs provide for two independent scavenge pumps, both requiring to be in use under normal working conditions. It is improbable that both can be damaged at one time, and seeing that the air supply in each cylinder contains three times the quantity of oxygen chemically necessary for the complete combustion of the fuel, it is expected that in the event of one pump only being at work there will be sufficient air supplied to either work all the cylinders with a reduced fuel supply or to work, say, three out of four of the cylinders at nearly full power.

It will not be out of place to discuss rules for determining suitable sizes of shafts for Diesel engines. In a steam engine it is usual to consider only the steam-pressure cylinder diameters, and stroke as affecting them. In other words, the shafts are proportioned to the torsion moment only, and the bending moment, which has to be borne by the crank shaft, is ignored, or, at most, is only indirectly allowed for by considering it to be proportional to the torsion moment. In a Diesel engine, however, the bending moment is the more important of the two, and demands to be taken into account.

It is not difficult to ascertain the torsion moment with precision, but the case is different in regard to the bending moment. Where a shaft is supported on two bearings only,

it is usual to consider the bending moment it sustains at any part to be equal to that which would be borne by a girder supported at the bearings and loaded at the position of the crank pin. This assumes that there is no bending moment at the bearings, and that the greatest bending comes upon the crank pin. Where, however, the crank shaft is supported at several bearings, the actual bending moments upon it depend upon the distribution of the supporting forces between the several bearings. It is generally considered that as regards bending moments the shaft is under the condition of a continuous girder, fixed in position, not merely supported, at each bearing. This assumption involves a bending moment at the bearings about equal in intensity, although of opposite direction, to that at the crank pin. If the load on the crank pin is assumed to be concentrated at its center, and the points of fixity of the bearings are at the centers of the journals, then the greatest bending moment upon a crank shaft will be given by

$M_b = \frac{1}{8}$th load $\times$ length between centers of bearings.

This is the amount of the bending moment, which has been taken in the following calculations. It is not put forward as being absolutely correct, but it is considered that it is sufficiently accurate to enable it to be used as a basis of comparison of the stresses in similar engines.

If the bending moment only were considered, then in engines of various proportions, assuming the same initial pressure and fuel supply, a rule

$$d \propto \sqrt{D^2 L}$$

would be suitable. If we consider torsion moments only, the rule would be

$$d \propto \sqrt[3]{D^2 S}$$

where

d = diameter of crank shaft.
D = diameter of cylinder.
S = stroke of piston.
L = space between centers of bearings.

As, however, both moments must be considered, it can be shown that a suitable rule can be made of the form

$$d = \sqrt[3]{D^3(A \times S + B \times L)},$$

where A and B are co-efficients whose proportion to one another depends upon the relative magnitudes of the bending and torsion moments, and whose actual amounts depend upon the working factor of stress deemed to be suitable.

In the comparisons which will be made it is assumed that it is desirable to have the same margin of security in the shafts of marine Diesel engines as experience has shown to be necessary in those of marine steam engines. For these latter the rules of Lloyd's Register have been taken as fixing a minimum size.

In determining the load which is actually exerted at any moment upon the crank pin, reference is made to the indicator diagram which shows the pressure exerted on the piston. Allowance then has to be made for the effect of the inertia of the reciprocating parts and for the obliquity of the connecting rod.

The effect of the inertia is to render the actual load on the crank pin less than that on the piston during the first part and greater during the latter part of the stroke. The amount of this difference between piston and crank-pin loads is negligible when the engines are running slowly, but is considerable with high speeds. At about the middle of the stroke there is no difference at any speed, and as it is at this part of the stroke that the leverage of the crank is greatest, it is found that the inertia effect is not so important when dealing with torsion moments as it is when bending moments are concerned, seeing that the latter do not depend upon the angular position of the crank at all.

As regards the rotatory motion of the engine as a whole, where there are two cranks at right angles, if the reciprocating weights upon each are equal, the effect of the inertia on one crank almost exactly balances that on the other, so that

the estimated combined torsion moment has the same value, whether the effect of inertia is taken into account or not. If there are three cranks equally spaced at 120 degrees apart and equally loaded, the combined effect of inertia is also very small, although not quite so small as with two cranks.

With four cranks equally spaced the effect is that of two pairs at right angles, and is therefore also negligible. In general where several cylinders are employed, the combined torsion moment will show the same result, whether the inertia is included or not.

Seeing that all engines have to be started and stopped, and that from the time they are started they have to progress through all ranges of speed from very slow up to full speed, it is necessary in their design to consider all the conditions under which they work. From what has been said, it is clear that if we neglect the inertia and provide for the conditions at starting those occurring at full speed will be less onerous and will therefore also be fully met.

In the comparisons of Diesel engine shafts with those of steam engines the inertia effect has, therefore, not been taken into account in either case, it being considered that by taking the most severe conditions in both cases a proper comparison will result.

For the Diesel engines with varying numbers of cylinders the particulars of piston loads taken are from data kindly supplied by Messrs. Sulzer Brothers relating to a marine engine made by them to work on the two-stroke cycle single acting. The indicator diagram is an actual one taken while full power was being developed. The initial pressure is 500 pounds per square inch, the mean forward pressure is 171.5 pounds per square inch, the mean pressure during compression is 55.5 pounds per square inch. These figures give a mean effective pressure of 116 pounds per square inch.

Fig. 1 shows the indicator diagram.

Fig. 2 shows the torsion moment which would occur at every position of the crank throughout one revolution in an engine with one cylinder only.

This diagram is important, as being the basis of a series of diagrams which were obtained by superimposing diagrams like Fig. 2 upon one another with the starting points in the positions appropriate to the positions of the cranks.

This series included the cases of two, three, four, five, six and eight cylinders respectively, and from them the results given in Tables I, II and III have been obtained. When the whole of the cylinders are thus taken account of the combined diagrams give only the torsion moments on the shafting abaft the aftermost crank pin, and thus apply only to the aftermost journal of the crank shaft and to the thrust, tunnel and screw shafting.

To determine the torsion moments on the other journals of the crank shaft a similar procedure has been taken. As an example of the method the case of a five-crank engine is illustrated in Figs. 7, 8 and 9.

If we consider the forward crank shaft first, it is seen at once that this part of the crank shaft is precisely in the condition of the shaft of a single-cylinder engine as shown in Fig. 2. The conditions of the other crank shafts will depend upon the sequence in which the cranks are arranged. It is probable that the arrangement which will best meet the requirements of this kind of engine is that shown in Fig. 6, the figures denoting the numbers of the separate cranks counting from forwards. In this arrangement each crank is as nearly as possible opposite to its next neighbors.

With this arrangement Fig. 7 shows the torsion moment on the aft journal of the second length of crank shaft, Fig. 8 that on the similar journal of the third length, and Fig 9 that on the fourth. From Fig. 7 is seen that in the second length the maximum positive and negative torsion moments are the same in amounts as in the first length of shaft. Fig. 8 shows that on the third length there is a maximum torsion 18 per cent. greater than that on the first shaft, but there is a less negative moment. Fig. 9 shows that on the fourth shaft there is the same maximum tension as on the third, with a slightly greater negative moment.

TABLE SHOWING THE VALUES OF THE VARIOUS MAXIMUM TORSION, BENDING AND COMBINED MOMENTS COMING UPON THE CRANK SHAFTS OF DIESEL ENGINES, THE PRODUCT OF PISTON LOAD AND STROKE BEING TAKEN AS 1.00 AND INERTIA BEING NEGLECTED. THE SPACE BETWEEN CENTERS OF BEARINGS IS TAKEN AS 1.2 TIMES THE STROKE.

No. cylinders.	Tunnel-shaft torsion moment.		Crank shaft under full load.			Crank shaft allowing for power absorbed by compressors.			Maximum torsion moment on tunnel shaft, allowing for power absorbed by compressors.
	Maximum.	Mean.	Maximum torsion moment.	Bending moment at instant of max. torsion.	Equiv. max. moment estimated as bending.	Maximum torsion moment.	Bending moment at instant of max. torsion.	Equiv. max. moment estimated as bending.	
	I.—DATA REFERRING TO AN ENGINE WITH MEAN EFFECTIVE PRESSURE OF 116 POUNDS PER SQUARE INCH.								
1	.300	.037	.300	.124	.254	.296	.124	.252	.296
2	.300	.074	.300	.124	.254	.293	.124	.250	.293
3	.261	.111	.300	.124	.254	.288	.124	.247	.250
4	.261	.148	.330	.124	.272	.315	.124	.263	.246
5	.250	.185	.354	.124	.287	.336	.124	.276	.232
6	.261	.222	.362	.124	.292	.340	.124	.278	.239
8	.320	.296	.420	.124	.328	.390	.124	.309	.291
	II.—DATA REFERRING TO AN ENGINE WITH MEAN EFFECTIVE PRESSURE OF 94 POUNDS PER SQUARE INCH.								
1	.262	.033	.262	.124	.232	.259	.124	.230	.259
2	.262	.065	.262	.124	.232	.255	.124	.228	.256
3	.228	.097	.262	.124	.232	.252	.124	.226	.218
4	.228	.129	.288	.124	.247	.275	.124	.239	.215
5	.219	.162	.309	.124	.260	.293	.124	.250	.203
6	.228	.194	.317	.124	.265	.298	.124	.253	.211
8	.280	.259	.367	.124	.295	.341	.124	.279	.254
	III.—MEAN OF THE ABOVE.								
1	.281	.035	.281	.124	.243	.278	.124	.241	.278
2	.281	.070	.281	.124	.243	.274	.124	.239	.275
3	.245	.104	.281	.124	.243	.270	.124	.237	.234
4	.245	.139	.309	.124	.260	.295	.124	.251	.231
5	.235	.174	.332	.124	.274	.315	.124	.263	.218
6	.245	.208	.340	.124	.279	.319	.124	.266	.225
8	.300	.278	.394	.124	.312	.366	.124	.284	.273

The figures given in the tables show that, whereas a progressive increase in the number of cylinders from 1 to 5 actually decreases the maximum stress on the tunnel and screw shafts, each increase in the number of cylinders above three produces an increase of the stress upon the crank shaft.

It is seen from Fig. 2 that the maximum torsion moment is three-tenths of the product of the piston load and the stroke. In Table I the values of the torsion and other moments are given in this proportion, the product of the piston load and stroke being taken as unity.

When the bending moment is considered it is found that the greatest intensity does not occur at the same instant as the maximum torsion stress. If we consider the case of an engine with the distance between the centers of bearings equal to 1.2 times the stroke, the maximum bending moment, which occurs when the crank is on the center will be .15 $\times$ load $\times$ stroke. The bending moment, however, which has to be considered is that which occurs at the same instant as the greatest torsion moment, and it is only .825 time that of the maximum amounts.

In determining the effects of the torsion and bending moments combined, the usual formula has been taken, namely—

$$M_c = .35\,M_b + .65\sqrt{M_b^2 + M_t^2}.$$

where

M_c is the combined moment estimated as bending moment.

M_b is the actual bending moment.

M_t is the actual torsion moment occurring at the same instant.

The values obtained for M_c are given in the tables. They will enable the size of the crank shaft to be determined, provided a definite value for the intensity of stress is agreed upon.

A similar set of calculations has been made, using diagrams obtained when working at ordinary full power in an engine made by another maker. These diagrams do not show so large an amount of power as those previously referred to;

but as they represent good practice, the values obtained from them have been given in Table II.

It has been thought desirable to use the mean of the two results to compare with the mean of results obtained from marine engines. The mean results have been stated in Table III.

In the Diesel engine a considerable amount of compressed air has to be provided for the injection of the fuel, and in the case of the two-stroke cycle engine the scavenge air also has to be compressed. The power required for these operations is usually obtained from the forward end of the crank shaft, but sometimes the various compressors are worked by levers actuated by the motion of one of the pistons. Where the power is obtained from the forward end of the crank shaft, the arrangement lessens the torsion moment upon all the shafting by an amount equal to that absorbed by the compressor. If it is assumed that 10 per cent. of the gross power is used in this way, and that the compressors are so designed as to use a fairly consistent torsion moment, the resulting effects on the shafts of the various engines are included in Tables I, II and III.

In order to determine a proper value of the intensity of stress to be used for crank shafts for marine Diesel engines, so as to preserve the same factor of security as that which obtains with the shafts of steam engines doing similar work, the particulars of seventeen engines by eleven different makers of repute were taken; these are given in Table IV. The torsion moment diagrams have been calculated for three of the engines—namely, Nos. 3, 7 and 11—and in these cases the ratios of maximum to mean stress were found to be 1.24, 1.28 and 1.265, giving an average value of 1.262. This has been assumed to be a reasonable value to take in the calculations of all the engines.

From the diagrams which were made it was apparent that, if the sum of the torsion moments of the high pressure and intermediate pressure only were considered, their combined effect had at no instant a maximum greater than the maxi-

TABLE IV.—GIVING PARTICULARS OF SHAFTS, ETC., OF MARINE ENGINES.

No.	Diameter of cylinders. Inches.	Pressure. Lbs. per sq. in.	Revolutions per minute.	Indicated horsepower.	Maximum steam pressure on pistons. Lbs. per sq. in.			Proportional loads on pistons.			Crankshaft diameter by L.R. rule.	Distance between centers of bearings. Inches.	Max. torsion moment, 1,000 lbs. per sq. in.	Max. bending moment, 1,000 lbs. per sq. in.	Max. torsion stress = 1.262 mean. Lbs. per sq. in.	Stress per combined moment. Lbs. per sq. in.
					H.P.	I.P.	L.P.	H.P.	I.P.	L.P.						
1	25, 40, 68 by 48 stroke.....	180	86	2,470	120	54	19.0	120	138	140.6	13.277	44	2,286	380	4,973	7,132
2	23½, 38, 64 by 42 stroke.....	180	67	1,353	124	45	16.0	124	117.5	118.4	12.198	39½	1,605	253	4,504	6,420
3	18, 30, 50 by 36 stroke.....	180	83	882.3	105	48.5	18.2	105	135	141	9.777	34	846	152	4,616	6,677
4	26, 43, 70 by 45 stroke.....	180	64	1,843	115	48	19.2	115	131	139	13.393	44½	2,291	411	4,857	7,024
5	30, 48, 78 by 54 stroke	160	62	2,882	100	54	18.0	100	137	121.5	14.932	57½	3,699	621	5,659	8,121
6	30, 48, 78 by 54 stroke.....	160	61	2,974	106	62	15.8	106	158.5	107	14.932	57½	3,880	545	5,935	8,361
7	25, 42, 71 by 48 stroke.....	200	78	2,821	128	58	18.5	128	163.5	148.5	13,894	52½	2,878	483	5,465	7,854
8	26½, 44, 72 by 48 stroke.....	180	76	2,770	120	57	17.0	120	157	125	13.876	49½	2,900	426	5,527	7,822
9	26, 43, 70 by 48 stroke.....	180	69	2,184	108	48.6	16.0	108	133	116	13,686	48½	2,519	371	5,004	7,085
10	24½, 40, 66 by 45 stroke.....	180	60.5	1,521	112	40.5	15.0	112	108.5	109	12,846	43½	2,001	278	4,807	6,764
11	27½, 43½, 73 by 48 stroke....	180	71	2,332	128	54.4	17.4	128	136	124	14.086	48	2,614	436	4,763	6,830
12	19, 31, 51 by 36 stroke.....	180	78	1,080	108	55	20.4	108	146	147	10.067	32½	1,102	168	5,500	7,796
13	26, 42, 70 by 48 stroke.....	180	62.5	1,991	121	44.7	19.5	121	116	141	13,573	45½	2,535	427	5,163	7,413
14	26, 42, 70 by 48 stroke.....	180	65	2,018	122	48	18.0	122	125	130	13.573	45½	2,471	394	5,032	7,181
15	25, 41, 68 by 48 stroke.....	180	62	1,632	117	48	15.1	117	129	112	13.330	48½	2,095	334	4,504	6,415
16	25½, 42, 69 by 48 stroke.....	180	70	2,276.5	118	54 5	17.75	118	148	129	13.508	45½	2,588	379	5,347	7,565
17	25, 42, 69 by 48 stroke..... .	180	75	2,420	114	44.8	16 o	114	126	122	13.395	45½	2,567	342	5,438	7,668
														Mean:	5,123	7,302

mum of the combined moment of the three engines. In other words, the maximum of the two engines does not occur near the points where the low-pressure moment is negative. It was also found that, even allowing for the somewhat greater load, which, in two cases out of the three, came upon the intermediate-pressure piston than upon the low-pressure piston, the effect of the combined bending of torsion moment upon the intermediate crank shaft always fell short of the combined effect on the low-pressure crank shaft. In the case of steam engines, therefore, the aftermost crank shaft is somewhat more stressed than either of the others, and it is therefore this shaft only which has been dealt with.

In making these calculations of the stresses caused by the bending moments and torsion moments combined the former have been assumed to be due to the greatest load coming on the piston which occurs at the commencement of the stroke. Although the load is slightly lessened at the instant when the greatest torsion moment is acting, the reduction is not nearly so much as that which occurs in the Diesel engine, and it has not been taken into account. In all these cases the size of shaft is assumed to be that which would be the minimum required by Lloyd's Rules, not the somewhat larger shafts actually fitted by the makers in each case.

The combination of the bending and torsion moments has been made by the usual formula, and as a result an average stress of 7,302 pounds per square inch was obtained. It must be distinctly stated again that it is not claimed that these figures are absolutely correct, as they are obtained upon certain assumptions regarding the bending moment; but they are reliable as a means of comparison with figures obtained in precisely the same way with other engines.

It now remains to apply the stress intensity to the shafts of the Diesel engine.

As an example, let us take an engine with, say, four cylinders, two-stroke cycle, single acting. Its diameter of cylinder $= D$ inches, its stroke S inches, and its span of bearings $L = 1.2\ S$; that is to say, its proportions being the same

as those taken in Tables I, II and III, and assume the compressors to be worked from the forward end of the crank shaft. We shall have the load on piston $= D^2 \times \frac{\pi}{4} \times 500$ pounds.

From Table III we see that the maximum combined moment on the crank shaft estimated as a bending moment is; Load on piston $\times$ stroke $\times$ .251. Hence if d is the diameter of shaft, which will give a stress of 7,300 pounds per square inch—

$$d^3 \times \frac{\pi}{32} \times 7{,}300 = D^2 \times \frac{\pi}{4} \times 500 \times S \times .251,$$

from which

$$d^3 = D^2 S \times .1375,$$

and

$$d = \sqrt[3]{D^2 S \times .1375}$$

This rule is only suitable for cases in which L is 1.28. As has been stated, the rule can be put into the form $D = \sqrt[3]{D^2(A S + B L)}$ to suit all ratios of $\frac{L}{S}$ from 1.2 to 1.8.

TABLE V.

Values of $(AS+BL)$ in the formula $D = \sqrt[3]{D^2 \times (AS+BL)}$.

Description of single-acting engine.	For engines with separate compressors.	For engines where compressors are worked from fore end of crank shaft.
1, 2 or 3 cylinders,	$.089 S + .037 L$	$.086 S + .038 L$
4 cylinders, . . .	$.099 S + .036 L$	$.093 S + .037 L$
5 or 6 cylinders, . .	$.111 S + .035 L$	$.103 S + .036 L$
8 cylinders, . . .	$.131 S + .033 L$	$.120 S + .034 L$

The above Table V gives the suitable values of $A S + B L$.

We have next to determine a rule for tunnel shafts, which are subject to torsion only. In the steam engine the usual practice is to make the tunnel shaft $\frac{20}{21}$ of the diameter of the crank shaft. In Table IV it is seen that the average stress

caused by the maximum torsion moment upon shafts of the sizes required by Lloyd's Rules for the crank shafts of the engines under consideration is 5,123 pounds per square inch. As the tunnel shafts will be $\frac{20}{21}$ of the sizes of the crank shafts, the average of the stresses on them will be

$$(\tfrac{21}{20})^3 \times 5{,}123 = 5{,}930 \text{ pounds per square inch.}$$

Taking again the four-cylinder engine, we see from Table III that the maximum torsion moment to be provided for is—

$$\text{Load} \times \text{stroke} \times .231$$

Hence, if d_t be the diameter of the tunnel shaft to give a maximum stress of 5,930 pounds per square inch, we get—

$$d_t \times \frac{\pi}{16} \times 5{,}930 = D^2 \times \frac{\pi}{4} \times 500 \times S \times .231$$

whence

$$d^3 = D^2 S \times .07791$$

TABLE VI.—GIVING THE VALUE OF THE COEFFICIENT REQUIRED IN THE FORMULA.

$$\text{Diameter of tunnel shaft} = \text{coefficient} \times \sqrt[3]{D^2 S}.$$

Description of single-acting engine.	For engines with separate compressors.	For engines where compressors are worked from fore end of crank shaft.
1 or 2 cylinders, . .	.456	.454
3, 4, 5 or 6 cylinders,	.436	.429
8 cylinders, . . .	.466	.452

In this way we get the above Table VI:

It is to be noted that no allowance is made for the equalizing effect on the torsion moment of the flywheel, which must be fitted in engines with a small number of cylinders. As against this, however, it may fairly be set that not only will

the flywheel be as small as will suffice to enable the engine to turn the centers, but also in these engines the range of stress through the inequalities of the torsion moments is considerably greater than in the steam engines, which are taken as standards of comparison, and experience shows that with a large range of stress a shaft cannot stand so high a maximum stress as may be safely borne when the range is small.

The calculations have been made for the two-stroke single-acting type of engine. They will all be equally applicable to the four-stroke single-acting engine with double the number of cylinders.

With regard to the double-acting type of two-stroke cycle engine, with the exception of there being no allowance made for the lessening of the piston area for the upstroke due to the piston rod, the single-cylinder engine will compare in all respects with the two-cylinder single-acting engine. The two-cylinder double-acting engine with cranks at 90 degrees will also compare with the four-cylinder single-acting so far as tunnel and screw shafts are concerned, but the crank shaft will not be so much strained, owing to there being no part of it in the same condition as the third crank of the four- cylinder engine. The maximum torsion on the crank shaft will be the same as that on the shaft of a two-cylinder single-acting engine. Three-cylinder engines with the cranks equally spaced, and four-cylinder engines with cranks spaced 90 degrees, 45 degrees, 90 degrees and 135 degrees will be under the same conditions as single-acting engines with six cranks at 60 degrees, and with eight cranks at 45 degrees respectively, there being in these cases the corresponding phases which bring the greater torsion moments upon some of the cranks.

The following additional paper will also prove of interest to such of our readers as are interested in the development of the marine Diesel engine:

MODERN DIESEL OIL ENGINES.

By F. Schubeler.

Read at the Zürich meeting of the Institution of Mechanical Engineers, July, 1911.

The practical development of the Diesel engine may be taken as having started shortly after Mr. H. Ade Clark read his paper* on "Diesel Engines" before the Institution of Mechanical Engineers in 1903.

As is already known, it was the intention of the inventor to burn coal direct in the working cylinder, which, however, proved to be impossible, as the unburned residues could not be removed from the latter. This condition of things will remain unchanged, and only oils will be used as fuel; these, however, being of the most composite properties.

For dealing with liquid fuel the propositions made by different designers varied principally only in the arrangement of the air pump and the construction of the fuel valve. For instance, Haselwander and Trinkler used the working cylinder, so to speak, as the first stage of the air pump by extracting air during the compression stroke from the former as soon as a certain pressure (about 100 pounds) was reached, and only producing the differential pressure for the injection air (700-900 pounds) in the actual air pump. Haselwander had his air pump combined direct with the working piston by a differential arrangement, whereas Trinkler provided a special air pump in the cylinder cover. Other designers proposed an intermediate process between those of the ordinary gas engine and the Diesel engine. Thus Sabathé advocated compression up to about 300 pounds, and artificial ignition of part of the fuel. The Brons motor, as still carried out by the Gasmotoren-Fabrik Deutz for smaller units, has also similar features. Generally, however, it can be stated that most of the above-mentioned propositions have disappeared and some have not even been carried out, all of them giving place to the well-known Diesel principle, with its characteristics: compression in the working

* Proceedings, I Mech. E., 1903, Part 3, Page 395.

cylinder up to the ignition temperature of the fuel (about 500 pounds or about 1,000 degrees F.), and creation of the pressure of the injection air (600-850 pounds per square inch), by means of which the fuel is introduced atomized into the cylinder in an independent multiple-stage air pump. It is only natural that the development of this principle has caused some difficulties in the beginning. In the hitherto known steam engine the working process takes place partly in the boiler and only partially in the working cylinder. The boiler, which has no moving parts, deals with the heaviest part of the process and furnishes the working fluid in a refined state to the working cylinder, whereas with the Diesel engine the whole process has to be dealt with in the working cylinder. That it has been possible within a respectably short period to develop the Diesel engine to such a degree as to enable it to compete successfully with the steam engine also for large units certainly proves the present-day skill and thoroughness of engineering. It may be stated that as regards reliability the Diesel oil engine can now be considered as fully equal to the steam engine, it being only a question of economy in each individual case which of the two prime movers takes the preference. There was one principal difficulty which had to be overcome. The extreme high pressure and temperatures of the Diesel process put a limit to the dimensions of the cylinders, which will hardly exceed a diameter of 30 inches. This corresponds to an approximate cylinder output of 300-400 H.P., assuming 150 revolutions per minute of the shaft and ordinary Otto cycle. Theoretically there would be nothing in the way to obtain any desired output by arranging any number of cylinders side by side. Practically, however, it is not advisable to provide for more than six cranks, as a larger number would produce an inadmissible twisting of the shaft. Furthermore, such units require a great amount of space, and would become extremely costly. These reasons called forth the necessity of seeking other means by which to increase the specific cylinder output of the ordinary single-acting Otto cycle engine.

For this there are the following possibilities:

(1) By carrying out the single-acting Otto cycle machine as a double-acting one.

(2) By adopting the single-acting two-stroke cycle working process.

(3) By adopting the double-acting two-stroke cycle working process.

The double-acting Otto cycle cylinder under (1) corresponds approximately to a doubling of the cylinder output.

Almost the same result can be obtained by the procedure under (2), which, however, necessitates, as will be explained later on, the provision of special scavenging pumps.

The solution under (3) would theoretically even quadruplicate (in fact, about 3.4) the output of the ordinary Diesel engine cylinder, and mean the adoption of the usual working process of steam engines, also for the oil engine.

The author will try in the following pages to throw a little more light on these various solutions in order that their respective advantages and disadvantages may be seen. As is known, this subject is for the time being a very disputed one, and has not developed so far yet as to permit any definite statement with regard to which of the solutions will be ultimately adopted for the large Diesel units. It has, however, been ascertained that for smaller and medium-sized units the single-acting Otto cycle takes the preference, the same involving the smallest working strain on the cylinder. Although this type of engine necessitates a more or less frequent cleaning of the internal parts, especially of the exhaust valves, the author is informed that such machines have worked for periods of six to eight weeks without interruption, even in cement factories and mills.

The normal type of the single-acting four-stroke cycle engine is a vertical one with trunk piston and without special crosshead. This has not only the advantage that the wear and tear of the piston is very slight, but also others with regard to the construction of the fuel valve, and the symmetric shape of the combustion chamber. The space required is comparatively small, which is frequently today an important advantage, and

permits a good utilization of the foundations upon which the enormous free forces of the cylinders are transmitted vertically.

The horizontal type, which is carried out by various firms, is specially suitable where small heights are essential, and resembles in its construction that of the normal gas engine. The free forces are more awkward to deal with, and some difficulty also arises with regard to the formation of the combustion chamber and the introduction of the fuel into the cylinder.

The Single-acting Two-stroke Cycle Engine.—The author will proceed first with the single-acting two-stroke cycle engine, and will deal with the double-acting engines later on. The first trials with a Diesel engine of this type were made some ten years ago. Owing perhaps to the choice of too small an output, and also some other inadequacies, this first attempt was not repeated for some time.

It may be useful to recall some of its principal features. An exhaust valve is dispensed with, the piston uncovering, at the end of its outward or downward stroke, slots arranged in the cylinder-wall through which the burnt gases escape from the cylinder. By means of a special scavenging pump, an air pressure of about 4 pounds is produced. This lightly-compressed air enters the cylinder through a valve arranged in the cylinder cover, drives the burnt gas out of the cylinder, and fills the latter again with a new charge of air as soon as the piston in its upward motion covers the slots. The air valve being closed, the air is compressed as in the ordinary Otto cycle engine, and also the fuel introduced in the same way at the end of the upward stroke. The whole process of driving out the gases and the refilling of the cylinder takes place within a very short time at the end of the outward or downward stroke. The length of the slots is about one-fifth of the stroke. With the ordinary Otto cycle machine two entire strokes are available for this process, giving, however, thus only half the number of impulses which can be obtained with the two-stroke cycle type.

As to the two-stroke cycle process, this is the same as the

well-known one employed with the highly developed gas engine (Körting, Oechelhäuser). It will be seen from the following that it is even more suitable for the Diesel engine. The two-stroke cycle gas engine scavenges also at the stroke end with fresh air, the difference, however, being that already at this stage of the process the combustible mixture is introduced into the cylinder by a special gas pump, and this even before the exhaust slots are entirely closed. It therefore cannot be avoided that part of the combustible gases escape unused in the exhaust, creating thereby a possible danger for the engine. With the Diesel engine the fuel is only introduced at the end of the compression stroke, thus preventing any losses of fuel and danger of pre-ignition. The Diesel engine has furthermore the advantage that the rather bulky and power-absorbing gas pump can be dispensed with. It has also been ascertained that two-cycle gas engines show an increased fuel consumption at partial loads, whereas the two-stroke cycle Diesel engine has also with smaller load consumptions, which only differ slightly from those obtained by normal load. With regard to the fuel consumption in general, it must of course be expected that the consumption of the two-cycle engine is somewhat larger with respect to the effective output than for the single-acting Otto cycle engine. This must amount at least to the percentage of the energy absorbed by the air pumps. There must also be another increase due to the short cleaning and filling process of the cylinders, which does not allow as perfect a scavenging as with the four-cycle engine. With the latter under its worst conditions about 7/10 per cent. of the cylinder volume (representing the compression space) is liable not to be fully replaced by fresh air.

Some Advantages.—As a main advantage it may be mentioned that the turning moment is much more favorable, and consequently the revolving masses for obtaining a certain degree of cyclic variation are much lighter. The two-cycle engine furthermore guarantees a better starting and better conditions for regulation, which is specially important for direct coupling with alternators. The space required is considerably

smaller, and the engine is also lighter and therefore cheaper. To give a figure, the difference in price between an ordinary single-acting four-cycle and a single-acting two-cycle engine for a capacity of 1,000 H.P. already amounts to about 25–30 per cent., which figure would have to be considerably increased if larger units were compared. Furthermore, the lubricating-oil consumption is somewhat smaller per horsepower, as the same cylinder and bearing dimensions are sufficient for producing about double the output.

It still remains for reference to be made to the chief advantage of the two-stroke engine—that is, the elimination of the exhaust valve (or two with larger units). Some difficulties cannot be avoided with the latter, as soon as the same attains larger dimensions. Generally speaking, the fewer working parts coming into contact with the hot gas the better. The scavenging valve (or valves) in the cylinder cover may be dispensed with. Only a part of the cylinder slots are then used as exhaust slots, the remainder on the opposite side being used for the introduction of the scavenging air. A special shape is given to the top of the piston to guide the current of the scavenging air and to guarantee a thorough removal of the burnt gases. The scavenging valve is thus removed from the high temperatures, and only the fuel valves and starting valves exposed to same.

An Attractive Problem.—It is self evident that for the continually advancing engineer the problem of the double-acting Diesel engine was attractive, especially as he already had excellent models in the gas engine, for which the stuffing-box problem may be regarded as solved. On the other hand, it must be admitted at once that with the Diesel engine the conditions to be dealt with are much more unfavorable, only to mention the higher pressure. It will, therefore, be preferable to avoid the difficulties with the stuffing box in the first instance, and to develop the new type on the lines of the single-acting four-stroke cycle engine, this all the more as the gain in output is practically the same. To go at once to the double-acting two-cycle engine without first gaining experience with

the single-acting engine seems to the author not to lie in the course of a natural development, and to be somewhat risky. As to the design of an engine of the latter type—namely, the double-acting two-stroke engine, this favors the provision of a special crosshead and also the horizontal arrangement. This being so, it may be derived from gas-engine and steam-engine practice that for the support of the pistons a rear stuffing box with bearing support for carrying the weight of the pistons has to be provided, which is certainly far from being desirable. In addition, great inconvenience is caused by the piston rod having to pass through the combustion chamber and necessitating an eccentric introduction of the fuel. The combustion process must therefore be far more unfavorable, and the construction of the valve gear more complicated. The proper setting of the two compression clearances on both sides of the piston is not as simple as with the single-acting engine, where by a shortening or lengthening of the connecting rod the compression can be proportioned to any amount desired. With the double-acting machine a compromise has always to be made, and the length of the piston rod has to be provided for an average compression on both sides.

When used for special purposes—the author has in mind its application for ship propulsion—the engine must be built as a vertical one, and it seems to him that the valves arranged in the lower cover of the cylinder are not easily accessible, and that also the difficulties with regard to the gear should not be overlooked. He would also mention the excessive height of this kind of machine, which is a great disadvantage when used for men-of-war, as there would be considerable difficulty in arranging such engines within the armored height.

Constructive Details.—The constructive details of the various types of Diesel engines may be described together, as a large number of the component parts remain the same whether used for this type of Diesel engine or the other. This especially applies to the fuel pump, fuel valve, air pump, piston, etc. The author will not, however, spend much time on this subject, as it is already sufficiently known, and will be elucidated

at Winterthur. The air pumps are today generally carried out as two or three-stage piston pumps with inter-coolers. Whereas smaller units have as a rule an air pump attached to each cylinder, driven by a lever from the connecting rod, it is a general practice to provide for larger units one large common air pump producing the injection air for three or more cylinders. The pump may be arranged either horizontally or vertically at one end of the bed plate. This arrangement reduces the number of engine parts to be controlled and also the first cost. The valves are generally metallic ones, which alone are able to stand the high temperatures.

The design of the scavenging pump used with two-stroke cycle engine does not call for any special remarks. The air supply may be controlled and regulated by poppet or slide valves. A very neat combination of this scavenging pump with the air pump is carried out by Messrs. Sulzer Brothers, who use the crosshead of the scavenging pump as low-pressure stage for the injection pump, which he also hopes to be able to show in the works.

For lubricating the cylinders, special small oil pumps—that is, one for each cylinder—are provided. For bearings, connecting rods, etc., forced lubrication is generally advocated. The pump for the forced lubrication is driven direct from the shaft, and may be a piston or cog-wheel pump. The motors fitted with forced lubrication are totally enclosed, the dripping oil being gathered in the bed plate and sucked up again by the pump after having passed through filters and coolers. Oil holes are bored in the shaft and connecting rods, which take the oil from the bearings up to the respective crosshead pins and other places where lubrication is required. The cylinders are always cooled, the pistons only from a certain size upwards. The water is generally led to the engine by gravity and only under a small head.

Piston Cooling.—For piston cooling several solutions are possible. In most cases a telescopic arrangement is provided. The disadvantage of this solution is the necessity of providing a stuffing box to prevent leaking water getting into the plate

and mixing with the lubricating oil. With the stuffing box arranged in the piston it is impossible to tighten it when the engine is running. If the same is arranged unmovable on the frame, heavy arms have to be attached to the piston, which involve a considerable eccentric strain on the latter, and which therefore cannot be recommended. In other devices joints are made use of, which, however, are subject to similar disadvantages. A very simple arrangement for the piston cooling, which does not require any stuffing box and the working of which is absolutely reliable, deserves mentioning. With this arrangement the water does not entirely fill the cooling space of the piston, the water being only squirted against the highly heated surfaces and draining off through a pipe which surrounds the spray pipe. By this arrangement all additional pressures and strains are eliminated. Some firms use oil for piston cooling, which may have some advantages, but which, on the other hand, has the disadvantage of a smaller cooling effect owing to the smaller specific heat and the smaller coefficient of transmission of this liquid. If the oil is taken from the general oil system of the engine the cooling device for the oil has to be increased accordingly. The exhaust valves for smaller motors are not cooled. Some firms use no cooling device even for larger motors, which is an important constructive simplification, whereas other firms have the larger exhaust valves water cooled.

Regulation.—The author will now make a few remarks about the regulating process of the Diesel engine. In practically all types the fuel valve is operated through levers and rollers by cams fixed on a side shaft, which open the valve during 10–15 per cent. of the stroke, irrespective of load. The amount of fuel allowed by the regulator for any specified load accumulates around the fuel-valve spindle in a space filled with the highly compressed air, which at the opening of the spindle forces the oil through an atomizer into the cylinder. At smaller loads the regulator acts upon the fuel valve in such a way as to reduce its quantity. The fuel pump has a constant stroke, and the regulating of the fuel quantity takes place by the governor

acting upon the suction valve, through which part of the fuel thus flows back into the suction chamber. If a greatly reduced oil quantity is delivered to the fuel valve, or, in other words, at small loads, it may happen that no ignition is effected, probably on account of the great air surplus extinguishing the forming flame. To avoid this the air pressure has to be reduced at smaller loads, which is done by hand in the case of smaller engines. Should this regulation be forgotten, no danger arises from the engine, the result being only a decrease in the number of revolutions until the regulator again delivers a somewhat increased fuel quantity. For larger engines, and especially where the conditions for synchronizing are more stringent, the regulation of the injection air is done automatically by the regulator.

Choice of Fuel.—A question which is of the greatest interest both to the manufacturer and to the client is the choice of fuel. It is certainly very surprising that the results of chemical analyses of different fuels do not furnish exact information with regard to the suitability of the fuel for Diesel engines so that this can only be decided by the actual tests. It happens that oil of a certain composition is quite satisfactory, whereas another of an absolutely similar consistency is unsuitable. It may be stated here that there is a large selection of cheap liquids available, the most important being: crude mineral oils, mineral oil residues (crude naphtha, mazout); further, gas oils—that is, intermediate products of oil refineries from which the lightest constituents, such as benzine and petroleum, have been distilled—and the gas-oil tars of water-gas works. More recently it has been found possible to utilize the by-products of the distillation of coal and paraffin, tar, solar and paraffin oil. Explosions or danger of fire on account of the high flash point are eliminated, and any quantities of such fuel may be stored without danger and without being subject to the restrictions made by law.

Piston Speed.—The different working conditions of the Diesel oil engines naturally require different types. The piston speed is generally between 600 and 1,000 feet per minute, which

cannot be reduced below a certain limit, as otherwise the slightest leakage could prevent the compression attaining the required amount. This is especially important for marine Diesel engines when running the same dead slow. The piston speed can, however, be as low as 180–200 feet per minute for continual service.

The High-speed Four-stroke Cycle Engine.—For standard slow-speed single-acting four-cycle engines the number of revolutions lies between 150 and 300 at respective capacities of 1,000–1,500 H.P. These engines have the best fuel consumption, but have also the disadvantage that they are heavy and expensive. For this reason a single-acting multiple-cylinder high-speed four-stroke cycle engine was created, which is especially suitable for direct coupling to dynamos. The speed varies between 220 and 350 revolutions per minute at units varying between 1,000 and 100 H.P. These high-speed types, if suitably designed and carried out by first-class manufacturers, are just as reliable in service as the slow-speed types. They have, however, a slightly higher fuel consumption. On account of their smaller first cost they are especially suitable where the Diesel engine is intended as a stand-by. For very small auxiliary engines up to about 100 H.P. on board ship the speeds are still higher, and range from 400–600 revolutions per minute and even higher.

Single-acting two-stroke engines with outputs ranging between 700 and 3,000 H.P. are carried out with speeds of 160–140 revolutions per minute.

For ship propulsion there is a tendency to use lower speeds in order also to obtain a good propeller efficiency. To give some figures 100–350 revolutions per minute may be taken as limits for sets of several 1,000 H.P. down to 100 H.P.

The Marine Diesel Engine.—This approximately defines the application of the Diesel oil engine for various purposes, amongst which the last-mentioned for ship propulsion is at present of universal interest. Hardly ever before has so much been written and discussed about an engine type as at present of the marine Diesel engine. It may well be stated that this

is frequently done by firms who have not recognized and thoroughly studied the difficulties of the problem. Such a treatment of this question, though somewhat characteristic of modern times, is certainly an unsound one, and failures are sure to follow. This will no doubt affect the reputation of the Diesel engine for marine purposes, and retard its general application, which will especially be regretted by those firms who are really able to build reliable engines for this purpose. The author takes this opportunity of requesting engineers to initiate a development which is somewhat slower and thereby more natural, as this alone will guarantee a lasting success.

Following this course the normal four-cycle Diesel engine was first adapted for marine purposes, and there are several boats running with such engines. This type, however, having as already mentioned, a limited output and being moreover heavy, has the further disadvantage that the most essential feature of the marine engine—namely, the problem of direct reversibility—cannot easily be solved. For smaller crafts the propeller can be made reversible, or a special reversing gear can be fitted into the shaft similar to those used in motor cars. Another possibility is to produce with a normal non-reversible Diesel engine electricity in a generator and have the propeller coupled to an electric motor, which can easily be made reversible. This system, the so-called Delproposto system, which otherwise gives good results, is not very economical on account of the numerous transformations of energy. It is also rather complicated, expensive, and heavy. The first boat of this type is the steamer *Vandal,* on the Volga, carried out by Messrs. Nobel. A similar Diesel boat, the *Venoge,* was built by Sulzer Brothers in 1904 for the Lake of Geneva. The engine has an output of 100 H.P. Another inconvenience of the four-stroke cycle type is the presence of exhaust valves, which necessitate a more or less frequent cleaning.

These difficulties, including also the last-mentioned one with the exhaust valves, could be overcome by adopting the two-cycle engine. A number of boats fitted with such engines, with outputs up to about 1,000 H.P., are in satisfactory service.

The reversibility can easily be attained. It may be mentioned that Messrs. Sulzer Brothers have done the pioneer work here in exhibiting the first engine of this type at the Exhibition of Milan in 1906. For outputs below 1,000 H.P., these engines may be of the ordinary trunk-piston type, whereas for larger units crossheads should be provided. The appearance of such an engine does not differ considerably from that of the well-known steam marine engine. Some firms have already gone further, claiming to be able to carry out a double-acting two-stroke machine. Unfortunately the results obtained with such engines are unknown to the author, and a definite decision—whether single-acting or double-acting engines are most suitable for marine purposes—cannot be given for the time being. The author for his part feels somewhat doubtful whether success has already attended such efforts. Preferably a two-propeller arrangement is adopted, and with larger units the scavenging pumps will be separated from the propulsion engines and driven by independent non-reversible Diesel units.

The space required for marine Diesel engines is smaller than for steam plants, the difference amounting to about 30–50 per cent., and the saving in weight is also considerable, reaching as much as 50 per cent. With marine Diesel engines it is possible to attain a larger action radius for ships, which can be five to six times as large as that for ordinary steamboats. The taking on board of the fuel is also simpler, cleaner and quicker than the coaling of steam vessels. An advantage, which must not be overlooked, is the saving in fuel consumption through the absence of preliminary heating and stand-by losses.

Reference may here be made to the paper read by Mr. Shackleton at the Institution of Marine Engineers, which gives an interesting comparison between "steam" and "oil," and contains in addition noteworthy information about the marine Diesel engine. Another important point is that the Diesel engine can be started at any time and within the shortest period, no extensive preparations being required. The largest output per cylinder may be taken at present at about 500 H.P., but considering the great efforts being put forward 1,000 H.P.

cylinders will soon be no rarity. As soon as this is the case the marine Diesel engine will be in a position to compete successfully with the steam engine, and will certainly hold the field.

Other Applications.—Regarding other applications of the Diesel engine, attempts have been made to build also Diesel locomotives, but the author regrets not being in a position to furnish any information.

The adoption of the Diesel engine for motor cars and aeroplanes does not seem to be very promising on account of the high pressures and consequent heavy and expensive constructions, not only for the engine but also for the accessories.

One word is required as to the thermal conditions of the engines. The calorific efficiency, respective mechanical output, may attain 30–40 per cent.; 25–30 per cent. are lost in the cooling water, and the rest is contained in the exhaust gases. Of the latter heat about 20 per cent. can eventually be utilized for heating purposes, and in adding also the heated cooling water the total efficiency of a Diesel unit may go up to 80 per cent., which is certainly remarkable. Messrs. Sulzer Brothers have brought out a special waste-gas utilizer, which it is hoped may be shown in the works.

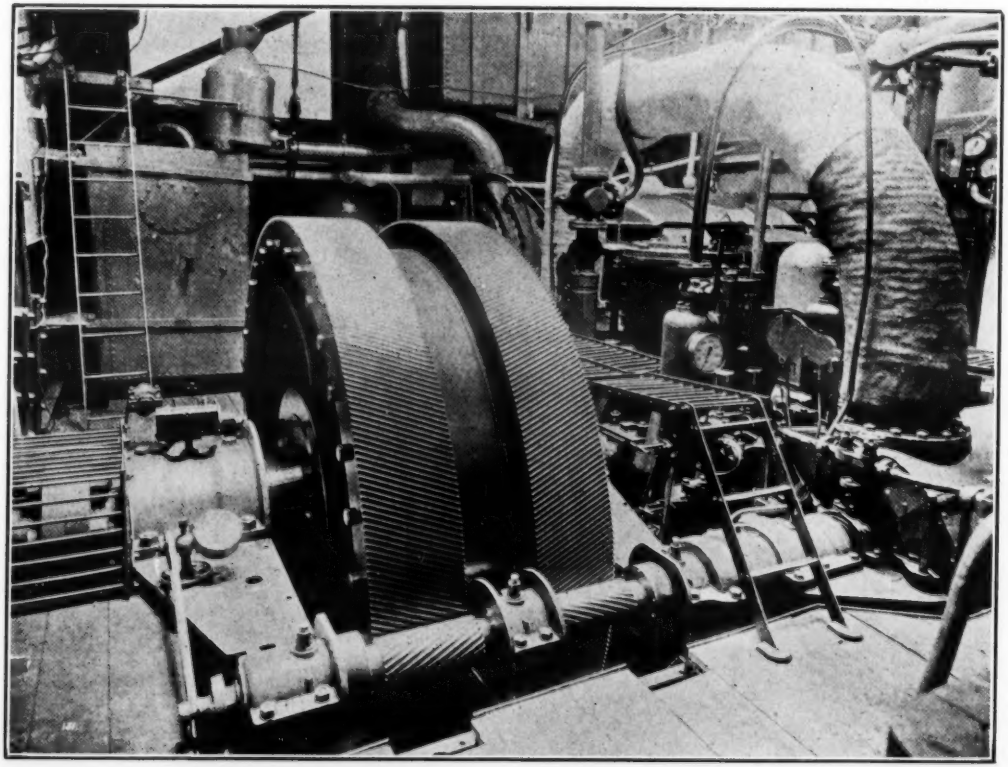
Cost of Running.—With regard to the actual cost per B.H.P. hour, in assuming a fuel price of 50s. per ton (which is certainly an outside figure) this will amount to about 0.11d. to 0.13d, or about 0.16d to 0.2d, per kw.-hour. These figures are the actual figures maintained during actual service, assuming, of course, that the engines are kept in good condition.

TWELVE MONTHS' EXPERIENCE WITH GEARED TURBINES IN THE CARGO STEAMER "VESPASIAN."

By the Hon. C. A. Parsons, C.B., LL.D., D.Sc., F.R.S., Vice-President, and R. J. Walker, Esq., Member.

Read at the Spring Meetings of the Fifty-second Session of the Institution of Naval Architects, April 5, 1911.

The object of the present paper is to lay before the members of the Institution a brief account of a year's working of the *Vespasian;* it may also be considered as a continuation of



ENGINE ROOM OF "VESPASIAN," SHOWING GEARING WITH COVER REMOVED.

the paper read at the Spring Meetings of last year on "The Marine Steam Turbine and Mechanical Gearing."

On completion of the trials referred to in last year's paper, the *Vespasian* was taken to sea on several occasions in light condition with a view to observing the behavior of the propelling machinery in a seaway. Although no very severe weather was encountered during these trials, sufficient evidence was obtained to show that the engines were not liable to race in heavy weather.

A series of careful observations was made of the variation in revolutions of the engines with the vessel pitching. The governors on the turbines were so adjusted as to come into operation when the revolutions of the engines exceeded 1,600, corresponding to 80 revolutions of the propeller shaft. Records were first taken at a normal rate of revolutions of 72 on the main shaft. The vessel was put head to sea, and during a period of pitching the revolutions of the propeller shaft reached a maximum of 82 at the moment when the stern of the vessel had reached the maximum angle of longitudinal inclination, the immersion of the propeller under this condition being about 6 inches to 12 inches. The governors of the turbines came into action just as the maximum angle of inclination was reached.

Further observations were made with the governors readjusted to higher revolutions, and the average records obtained showed a maximum variation in revolutions of from 72 to 84 during a period of pitching. The steam pressure at the H.P. turbine remained constant throughout at 70 pounds pressure.

As the result of these trials it was found that even with the governors out of action the acceleration of the engines during a period of pitching did not exceed 16 per cent. as a maximum, due to the very great angular momentum of the turbine.

In order to test the durability of the gearing the *Vespasian* was placed in regular service on June 9 last year, carrying coal from the Tyne to the Continent, and returning in water ballast. From that date to the present time she has completed twenty-six trips to Rotterdam and six trips to Antwerp, and during this period has steamed about 20,000 miles.

At various intervals the gear wheel and pinions have been inspected and tested for "back-lash," and only a month ago this was done, when practically no slackness in the teeth of the gearing could be detected. The wear in the teeth so far seems to be a negligible quantity.

We have here today for the inspection of the members of the Institution one of the actual pinions of the *Vespasian*, which has been in constant use since the new machinery was installed, the vessel having steamed over 18,000 miles prior to its removal.

On the first four voyages careful measurements of water consumption were taken. The following table gives the data and results obtained on these voyages:

Date	9-vi-10	16-vi-10	16-vi-10	22-vi-10	22-vi-10	22-vi-10	29-vi-10
Speed by log, knots..	9.35	9.22	10.58	9.61	9.27	10.22	9.37
Revolutions per min.	65.00	64.9	73.0	64.8	63.85	70.6	62.9
Boiler pressure, lbs. per sq. in.....	137	135	145	135	135	140	135
High-pressure turbine (initial pressure) lbs. per sq. in.	86	86	121	86	86	111	81
Vacuum (in inches).	28.5	29.1	28.6	28.55	28.4	28 4	28 3
Barometer	30.01	30.5	30.52	29.9	29.9	29.88	29.6
Water, main engines, lbs. per hour.....	12,140	12,300	15,680	11,890	11,730	14 510	11,100
Shaft horsepower.....	740	736	1,080	735	710	960	668
Water consumption, lbs. per S.H.P.....	16.4	18.0	14.5	16.2	16.5	15.1	16.6

The water consumptions per hour at the several rates of revolutions have been shown in Diagram 1 (Plate 1). The full line represents the data as obtained off the Tyne in the trials of March, last year, and referred to in the previous paper on this subject. The spots as indicated thus (.) show the data as obtained on service.

It will be noted that the service results agree very closely with the trial results. The displacement of the vessel on service when the data were taken was 4,560 tons, being about 5 per cent. greater than on the original trials.

In regard to the general behavior of the machinery it may be mentioned that during the winter months the vessel has

encountered some exceptionally severe weather, especially in light condition, and the turbines and gearing have worked throughout with entire satisfaction, and have not given the least trouble whatever.

A very noticeable feature has been the remarkable freedom from racing of the engines under conditions when the propeller has been entirely out of the water, confirming the observations made on the earlier trials when the engines, under the worst conditions, did not tend to accelerate in revolutions more than 16 per cent. The engineer of the vessel reports that without the aid of a sensitive tachometer it is very difficult to observe any acceleration in the speed of the engines.

On one occasion, namely, on October 12, when the vessel was returning in water ballast from Rotterdam, a very severe northeast gale was met with, necessitating increasing the steam pressure at the engines by about 25 per cent. to make headway against the heavy sea running.

With a view to experimenting with different qualities of steel additional pinions were made, one of chrome-nickel steel with a tensile strength of 55 tons, elastic limit of 38 tons and an elongation of 20 per cent. in a length of 2 inches, and two of special carbon steel of 40 to 45 tons' tensile strength, elastic limit 22½ tons and an elongation of 20 to 25 per cent. The original pinions were of mild chrome-nickel steel with a tensile strength of 37 to 38 tons and an elastic limit of 32 tons.

The chrome-nickel steel pinion of 55 tons' tensile strength was tried in August last, but was removed after two voyages, as on examination it was found to be fractured at the corners of some of the teeth. This appeared to be due to the teeth having been irregularly machined and bearing over one-third of their length only; the material also appeared to be of too brittle a nature. The original pinion was replaced and is still in use.

The pinion removed last month for the inspection of the members of the Institution has been replaced by one of the special carbon steel set of 40 to 45 tons' tensile strength, and is in use at the present time.

At the end of last year a new propeller was fitted to the vessel; this one was of the same diameter and blade area as the original propeller, but with a finer pitch ratio. The actual dimensions of the new propeller are as follows: Diameter, 14 feet; pitch, 14.14 feet, and expanded blade area, 72 square feet, as compared with the original propeller of 14 feet diameter, 16.35 feet pitch and 70 square feet expanded area.

The following table gives the data and results of a trial carried out off the Tyne on January 9 of this year, the vessel being loaded to the same displacement as on the original trial:

Speed in knots	9.31	9.66	9.94	10.34
Revolutions per minute	68.4	71.2	73.7	77.0
Boiler pressure, pounds	130	142	146	141
H.P. turbine, initial pressure	76	86	96	109
Vacuum (in inches)	28.8	28.6	28.55	28.4
Barometer	30.2	30.2	30.2	30.2
Water consumption, main engines, lbs. pr. hr.	10,400	11,510	12,590	14,000
Shaft horsepower	630	720	815	945
Water consumption, pounds per S.H.P. hour	16.5	15.98	15.45	14.81

The water consumption for the several rates of speed have been plotted on Diagram 2, shown in full lines for the new propeller and in dotted lines for the original propeller. Diagram 3 shows the revolutions obtained at the several rates of speed for each propeller, and Diagram 4 the shaft horsepower in terms of water consumption per hour. The full lines in each case are for the new propeller and the dotted lines for the original propeller.

It will be noted that at the speed of 9.3 knots, which is about the average service speed of the vessel, the water consumption per hour has been further reduced with the new propeller by about seven per cent., four per cent. being attributable to the increased efficiency of the turbines running at the higher revolutions, and the remaining three per cent. due to a slightly more efficient propeller.

In regard to the question of economy generally, as mentioned in the reply to the discussion on last year's paper, we do not profess that the results obtained with the *Vespasian* are the highest obtainable with geared turbines. In a new

Diagram 1.

N° 1 PROPELLER 14·0 DIA. 16·35 PITCH. EXP. AREA 70 SQ. FT.

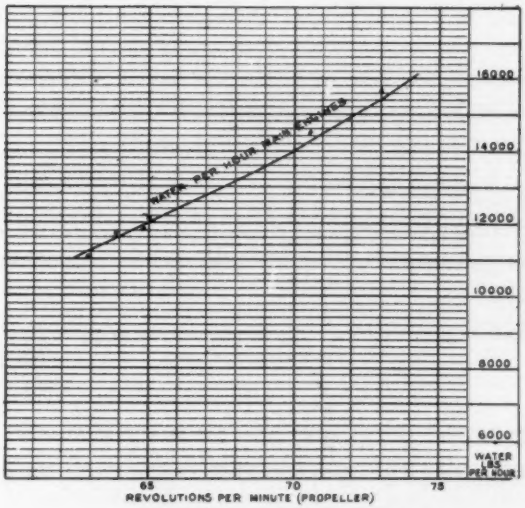


Diagram. 2.

N° 1 PROPELLER 14·0 DIA. 16·35 PITCH. EXP. AREA 70 SQ. FT.
N° 2 " " 14·0 DIA. 14·14 PITCH " " 72 " " "

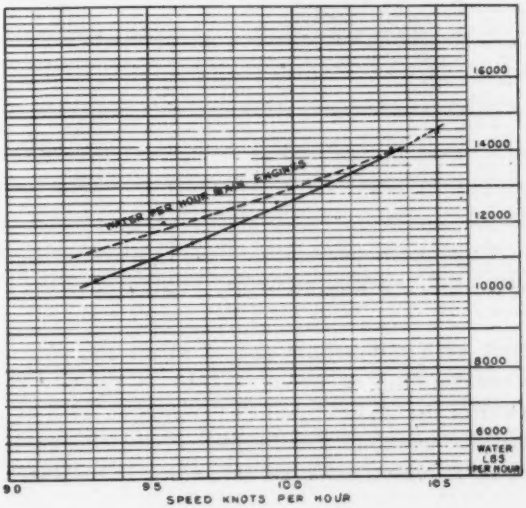


Diagram 3.

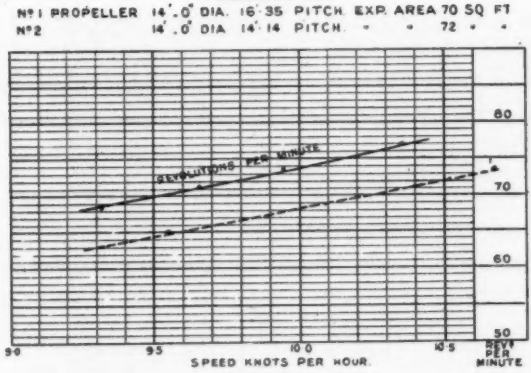
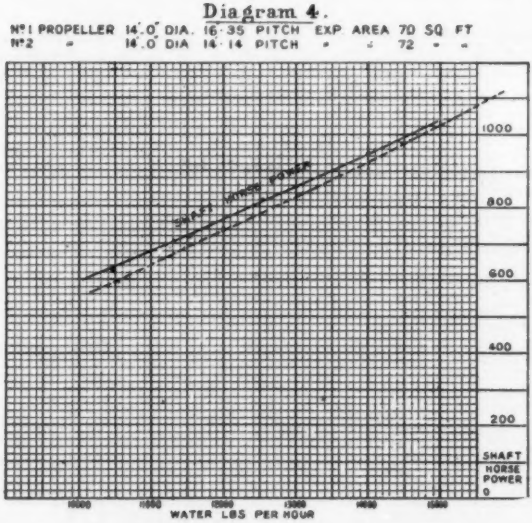


Diagram 4.



vessel, and with new boilers, say, of about 180 pounds' pressure, a consumption of about 12½ to 13 pounds per shaft horsepower of the main engines, as compared with 16 pounds at the service speed of the *Vespasian*, could confidently be expected in an installation of, say, 1,000 H.P., which would

be equivalent to 11½ to 12 pounds per indicated horsepower of reciprocating engines, assuming a ratio of shaft horsepower to indicated horsepower of 91 to 92 per cent.

In conclusion, it may be mentioned that the introduction of gearing into war vessels presents great advantages, since its adoption would result in increased economy, more especially at cruising speeds.

In a large war vessel, say, having four shafts, the gearing of the highest pressure portion of the turbine to the low-pressure shafts in lieu of direct-coupled cruising turbines, effects a saving in consumption of fully 25 per cent., at a cruising speed equal to one-tenth of the full power, and 30 per cent. at one-fifteenth of the full power, giving corresponding increases in the cruising radius with the same quantity of coal.

CANADIAN AND AUSTRALIAN NAVIES.

It would be difficult to exaggerate the importance of the agreements arrived at concerning the defences of the Empire by the recent Imperial Conference. The Parliamentary paper issued on Monday deals with the beginning of systematic naval and military coöperation by the various members of the Empire and is likely to prove historic in every sense of the word. According to the arrangements now made with the Dominions the naval services and forces of Canada and Australia will be exclusively under the control of their respective governments. The training and discipline of the naval forces of the Dominions will be generally uniform with the training and discipline of the fleet of the United Kingdom, and, by arrangement, officers and men of the said forces will be interchangeable with those under the control of the British Admiralty. The ships of each Dominion naval force will hoist at the stern the white ensign as the symbol of the authority of the Crown, and at the jack-staff the distinctive flag of the Dominion.

Naval Stations.—Connection with Admiralty.—The Cana-

dian and Australian governments will have their own naval stations. In the event of their desiring to send ships to a foreign port they will obtain the concurrence of the Imperial Government, so that the Foreign Office may make the necessary arrangements in the manner usual between the Foreign Office and the Admiralty. While Dominion ships are at a foreign port a report of their proceedings will be forwarded by the officer in command to the Commander-in-Chief on the station, or to the British Admiralty. The officer commanding a Dominion ship, so long as he remains in a foreign port, will obey any instructions he may receive from the British Government as to the conduct of any International matters that may arise, the Dominion Government being informed. When a ship of the British Admiralty meets a ship of the Dominions, the senior officer will have the right of command in matters of ceremony or international intercourse, or where united action is agreed upon, but will have no power to direct the movements of ships of the other service unless the ships are ordered to coöperate by mutual arrangement. In foreign ports the senior officer will take command, but not so as to interfere with the orders that the junior may have received from his own Government. The British Admiralty undertakes to lend to the Dominions during the period of development of their services, under conditions to be agreed upon, such flag officer and other officers and men as may be needed.

Training. Control in Time of War.—Important provisions relate to the question of joint training and to the procedure which would be followed in the event of war. It is considered desirable, in the interests of efficiency, and coöperation, that arrangements should be made from time to time between the British Admiralty and the Dominion for the ships of the Dominion to take part in fleet exercises or for any other joint training considered necessary under the senior naval officer. While so employed, the ships will be under the command of that officer, who would not, however, interfere in the internal economy of ships of another service further than absolutely necessary. In time of war, when the naval service of a Do-

minion, or any part thereof, has been put at the disposal of the Imperial Government by the Dominion authorities, the ships will form an integral part of the British fleet, and will remain under the control of the British Admiralty during the continuance of the war. The Dominion having applied to their naval forces the King's relations and Admiralty instructions and the Naval Discipline Act, the British Admiralty and Dominion Governments will communicate to each other any changes which they propose to make in those regulations or that Act. There are many other important clauses in this interesting agreement in the carrying out of which we imagine wireless telegraphy will play no inconsiderable part.

Conditions in Canada and Australia Compared.—The broad principles upon which the constitution and control of both the Canadian and Australian Naval units is based are practically identical, but, says the "Morning Post," they differ in many important details. For instance, in Australia the principle of compulsory training appears to be applicable to both military and naval service. In Canada, under the Military Act, the whole population between the ages of eighteen and sixty years is liable to military service, but the Naval Act differs from the Militia Act in the fact that service under the former is voluntary. The Canadian ships are to be built in Canada, but Australia's first fleet is being built in Great Britain; subsequent ships will probably be built in Australia. The Canadian unit, being intended for the defence of a double seaboard—the Atlantic and the Pacific—will be greater than the Australian with only one continuous seaboard to defend. In this connection it is interesting to note that the Canadian Premier estimates that the British cost of building will be exceeded by 33 per cent. if the work is carried out in Canada; the Australian Minister estimates that the cost of building in the Commonwealth will exceed the British figures by 25 per cent.—*"Page's Weekly."*

SHIPS.

UNITED STATES.

Torpedo-Boat Destroyers Nos. 43 to 50. Proposals for Construction.—Bids were opened August 7, 1911, for the construction of the above-named vessels. The particulars of the bids are given in the accompanying table. The contracts have not yet been awarded.

Collier Neptune.—This vessel, which is fitted with Westinghouse turbines and the Melville-Macalpine reduction gear, was given her preliminary contract trials July 26–29, 1911. On the forty-eight-hour full-speed and endurance trial the contract speed of 14 knots was not maintained, due to the use of very inefficient screws. The average horsepower was 5,409 B.H.P., corresponding to 5,879 I.H.P. The mean revolutions were 119.5 and the mean speed was 12.926 knots. The coal consumption was 1.79 pounds per I.H.P. per hour.

The reduction gear worked very satisfactorily on the trials, with but little noise and no appreciable vibration.

New propellers of more efficient design will be fitted, and better results may be anticipated in the near future.

Collier Jupiter. Electric System of Propulsion.—This vessel is a sister ship to the *Cyclops* and *Neptune* and is building at the Navy Yard, Mare Island, California. A contract has been entered into with the General Electric Company for machinery of the type described by Mr. W. L. R. Emmet in the February, 1911, number of the JOURNAL. The hulls of these three vessels are identical, and there will thus be afforded a means of direct comparison of the two new systems of propulsion with the reciprocating-engine installation in the *Cyclops*.

TORPEDO-BOAT DESTROYERS NOS. 43 TO 5

CLASS 1.—PARSON'S TURBINES, THREE SHAFTS, REC
CLASS 2.—PARSON'S TURBINES, THREE SHAFTS, CR
CLASS 3.—BIDDER'S DESIGN OF MACHINERY.

Bidders.	Class of bid.	Number of vessels to build.	Amount of bid for each vessel.	Time to build.	Speed, in knots.	Guaranteed maximum consumption of fuel oil per knot run.					Mean trial displacement.	Shaft H.P. for contract speed.	Length of vessel.	Length of machinery spaces.				Port shaft.
						29 knots.	24 knots.	20 knots.	15½ knots.	12 knots.				Boiler rooms.	Engine room.	Auxiliary room	Total.	
				mos.		*lbs.*	*lbs.*	*lbs.*	*lbs.*	*lbs.*			*f. i.*	*f. i.*	*f. i.*	*f. i.*	*f. i.*	
Bath Iron Works	1	2	761,500	24	29	710	440	280	220	175	1,010	16,000	300-0	73-6	40-3	12-3	126-0	1 L.P. turbine, 1 recip'g e
	3	2	741,500	24	29	710	440	280	220	175	1,010	16,000	300-0	70-0	36-9	10-6	117-3	1 L.P. turbine, 1 astern tu
New York Shipbuild'g Co.*	1	1	783,500	24	29	700	460	300	225	173	1,010	16,000	300-0	73-6	40-3	12-3	126-0	L.P. turb. and recip'g e
	2	1	776,500	24	29	700	460	270	210	170	1,010	16,000	300-0	73-6	40-3	12-3	126-0	H.P. cruising and L.P. t
	3	1	774,500	24	29	630	435	270	210	170	1,010	16,000	300-0	73-6	40-3	12-3	126-0	H.P. cruising and L.P. t
	3	1	761,500	24	29	700	460	305	210	175	1,073	15,900	300-0	73-6	40-3	12-3	126-0	Turbine reciprocating en
Fore River	3	1	774,950	24	29	663	377	240	184.5	151	1,010	15,350	300-0	73-6	38-6	14-0	126-0	Turbine
	3	1	779,450	24	29	686	402	269.5	141	110	1,010	15,350	300-0	73-6	38-6	14-0	126-0	Turbine and recip'g eng
Union Iron Works	1	2	803,650	24	29	680	420	290	225	170								
	2	2	807,500	24	29	680	420	290	225	170	...	...	...	...	...	...	...	1 L.P. turb., 1 cruising
Newport News Co.	1	1	820,000	24	29	685	510	338	252	217	1,010	15,400	300-0	73-6	40-3	12-3	126-0	Recip'g engine, L.P. a and astern turbine.
		1	801,000															
	2	1	823,000	24	29	685	445	315	225	205	1,010	15,400	300-0	73-6	40-3	12-3	126-0	H.P. cruising turbine, ahead and astern turbi
		1	804,000															
	3	1	815,000	24	29	700	465	320	220	195	1,010	15,400	300-0	73-6	38-6	14-0	126-0	Reciprocating engine, a and astern turbine.
		1	799,000															
	3	1	805,000	24	29	685	440	295	240	215	1,010	15,400	300-0	73.6	38-6	14-0	126-0	Ahead and astern turbin
		1	790,000															
Wm. Cramp & Sons†	1	1	770,000	24	29	710	475	320	218	175	1,036	...	300-0	73-6	40-3	12-3	126-0	1 L.P. turbine, 1 recip'g e
		2	760,000	1-23½, 1-24														
	1	3	758,000	1-23, 1-23½, 1-24	29	710	475	320	218	175	1,036	...	300-0	73-6	40-3	12-3	126-0	1 L.P. turbine, 1 recip'g e
	1	4	756,000	1-22½, 1-23, 1-23½, 1-24	29	710	475	320	218	175	1,036	...	300-0	73-6	40-3	12-3	126-0	1 L.P. turbine, 1 recip'g e
	3	1	775,000	24	29.5	705	445	315	213	170	1,046	16,500	300-0	73-6	40-3	12-3	126-0	1 turbine, 1 recip'g engin
		2	766,500	1-23½, 1-24														
	3	3	763,500	1-23, 1-23½, 1-24	29.5	705	445	315	213	170	1,046	16,500	300-0	73-6	40-3	12-3	126-0	1 turbine, 1 recip'g engin
	3	4	761,000	1-22½, 1-23, 1-23½, 1-24	29.5	705	445	315	213	170	1,046	16,500	300-0	73-6	40-3	12-3	126.0	1 turbine, 1 recip'g engin
	1	2	756,000	1-23½, 1-24	29	710	475	320	218	175	1.036	...	300-0	73-6	40-3	12-3	126-0	1 L.P. turbine, 1 recip'g e
	3	2	761,000	1-22½, 1-23	29 5	705	445	315	213	170	1,046	6,500	300-0	73-6	40-3	12-3	126.0	1 turbine, 1 recip'g engine

* If tried on Delaware Breakwater course, deduct $4,000.

S. 43 TO 50—CHARACTERISTICS OF PROPOSALS OPENED AUGUST 7, 1911.

E SHAFTS, RECIPROCATING CRUISING ENGINES, ON WING SHAFTS. DEPARTMENT DESIGN.
E SHAFTS, CRUISING TURBINES, WITH REDUCTION GEAR. DEPARTMENT ALTERNATE PROPOSAL.
INERY.

Arrangement of machinery.			Type of turbines.	Revolutions at designed power.	Steam pressure.		Reciprocating engines.			Operating through speed-reduction gear.	
Port shaft.	Center shaft.	Starboard shaft.			At boiler.	At engine.	Revolutions.	Cylinder diams.	Stroke.	Max. S.H.P.	Revolutions.
					p.s.	*p.s.*		*ins.*	*ins.*		
P. turbine, 1 recip'g engine.	1 H.P. turbine.....	1 L.P. turbine, 1 recip'g engine,	Parsons.								
P. turbine, 1 astern turbine.	1 H.P. turbine, 1 compound engine,	1 L.P. turbine, 1 astern turb....	Parsons.....	710	260	240	340 about	...	...	...	...
turb. and recip'g engine.	Main H.P. turbine..	L.P. turb. and recip'g engine.	Parsons.....	700	260	240	345	11½-17½	14	None	none
cruising and L.P. turb...	Main H.P. turbine..	I.P. cruising and L.P. turbine.	Parsons.....	700	260	240	None	None	none	8,000	550
cruising and L.P. turb...	Main H.P. turbine..	I.P. cruising and L.P. turbine.	Parsons.....	700	260	240	None	None	none	None	none
ine reciprocating engine...	.....	Turbine, reciprocating engine..	Curtis	550	265	240	350	12½-26½	13½	...	...
ine.....	.....	Turbine.....	Curtis	547	265	250	...	...	...	...	...
ine and recip'g engine....	.....	Turbine and recip'g engine.....	Curtis	547	265	250	953 at 75½ k.	12½-26½	13½	...	...
P. turb., 1 cruising turb...	1 H.P. turbine.....	1 L.P. turb., 1 cruising turb....	Main Parsons, cruising Westinghouse.								
p'g engine, L.P. ahead d astern turbine.	Main H.P. turbine...	Recip'g engine, L.P. ahead and astern turbine.	Parsons.....	710	260	240	345	11½-17½	14	...	...
cruising turbine, L.P. ead and astern turbine.	Main H.P. turbine..	I.P. cruising turbine, L.P. ahead and astern turbine.	Parsons.....	710	260	240	...	...	...	1,590	345
procating engine, ahead d astern turbine.	.....	Reciprocating engine ahead and astern turbine.	Curtis	550	265	250	260	12½-26½	13½	...	...
d and astern turbine.....	.....	Ahead and astern turbine.....	Curtis	...	265	250	...	...	...	...	...
P. turbine, 1 recip'g engine.	1 H.P. turbine.....	1 L.P. turb., 1 recip'g engine ..	Parsons.....	710	260	240	380	11-21	12	...	...
P. turbine, 1 recip'g engine.	1 H.P. turbine.....	1 L.P. turb., 1 recip'g engine...	Parsons.....	710	260	240	380	11-21	12	...	...
. turbine, 1 recip'g engine.	1 H.P. turbine.....	1 L.P. turb., 1 recip'g engine...	Parsons.....	710	260	240	380	11-21	12	...	...
bine, 1 recip'g engine.....	.....	1 turbine, 1 recip'g engine	Cramp	620	260	240	300	19-23	12	...	...
bine, 1 recip'g engine.....	.....	1 turbine, 1 recip'g engine	Cramp	620	260	240	300	19-23	12	...	...
bine, 1 recip'g engine.....	.....	1 turbine, 1 recip'g engine	Cramp	620	260	240	300	19-23	12	...	...
P. turbine, 1 recip'g engine	1 H.P. turbine.....	1 L.P. turb., 1 recip'g engine..	Parsons.....	710	260	240	380	11-21	12	...	...
bine, 1 recip'g engine.	.....	1 turbine, 1 recip'g engine	Cramp	620	260	240	300	19-23	12	...	...

† If tried on Delaware Breakwater course, deduct $4,900.

‡ O

RPEDO-BOAT DESTROYERS NOS. 43 TO 50—CHARACTERISTICS OF PROPOSALS OPENE

CLASS 1.—PARSON'S TURBINES, THREE SHAFTS, RECIPROCATING CRUISING ENGINES, ON WING SHAFTS. DEPARTMENT
CLASS 2.—PARSON'S TURBINES, THREE SHAFTS, CRUISING TURBINES, WITH REDUCTION GEAR. DEPARTMENT ALTERN
CLASS 3.—BIDDER'S DESIGN OF MACHINERY.

Maximum of fuel oil gun.		Mean trial displacement.	Shaft H.P. for contract speed.	Length of vessel.	Length of machinery spaces.				Arrangement of machinery.			Type of turbines.	Revolutions at designed power.	S pre
15½ knots.	12 knots.				Boiler rooms.	Engine room.	Auxiliary room	Total.	Port shaft.	Center shaft.	Starboard shaft.			At boiler.
lbs.	*lbs.*			*f. i.*	*f. i.*	*f. i.*	*f. i.*	*f. i.*						*p.s.*
220	175	1,010	16,000	300-0	73-6	40-3	12-3	126-0	1 L.P. turbine, 1 recip'g engine.	1 H.P. turbine.....	1 L.P. turbine, 1 recip'g engine,	Parsons.		
220	175	1,010	16,000	300-0	70-0	36-9	10-6	117-3	1 L.P. turbine, 1 astern turbine.	1 H.P. turbine, 1 compound engine,	1 L.P. turbine, 1 astern turb....	Parsons.....	710	260
225	173	1,010	16,000	300-0	73-6	40-3	12-3	126-0	L.P. turb. and recip'g engine.	Main H.P. turbine..	L.P. turb. and recip'g engine.	Parsons.....	700	260
210	170	1,010	16,000	300-0	73-6	40-3	12-3	126-0	H.P. cruising and L.P. turb...	Main H.P. turbine..	I.P. cruising and L.P. turbine.	Parsons.....	700	260
210	170	1,010	16,000	300-0	73-6	40-3	12-3	126-0	H.P. cruising and L.P. turb...	Main H.P. turbine..	I.P. cruising and L.P. turbine.	Parsons.....	700	260
210	175	1,073	15,900	300-0	73-6	40-3	12-3	126-0	Turbine reciprocating engine...	.....	Turbine, reciprocating engine..	Curtis	550	265
184.5	151	1,010	15,350	300-0	73-6	38-6	14-0	126-0	Turbine.....	.....	Turbine.....	Curtis	547	265
141	110	1,010	15,350	300-0	73-6	38-6	14-0	126-0	Turbine and recip'g engine....	.....	Turbine and recip'g engine.....	Curtis	547	265
225	170													
225	170	...	...	...	...	...	...	...	1 L.P. turb., 1 cruising turb...	1 H.P. turbine.....	1 L.P. turb., 1 cruising turb.....	Main Parsons, cruising Westinghouse.		
252	217	1,010	15,400	300-0	73-6	40-3	12-3	126-0	Recip'g engine, L.P. ahead and astern turbine.	Main H.P. turbine...	Recip'g engine, L.P. ahead and astern turbine.	Parsons.....	710	260
225	205	1,010	15,400	300-0	73-6	40-3	12-3	126-0	H.P. cruising turbine, L.P. ahead and astern turbine.	Main H.P. turbine..	I.P. cruising turbine, L.P. ahead and astern turbine.	Parsons.....	710	260
220	195	1,010	15,400	300-0	73-6	38-6	14-0	126-0	Reciprocating engine, ahead and astern turbine.	.....	Reciprocating engine ahead and astern turbine.	Curtis	550	265
240	215	1,010	15,400	300-0	73.6	38-6	14-0	126-0	Ahead and astern turbine.....	.....	Ahead and astern turbine.....	Curtis	...	265
218	175	1,036	...	300-0	73-6	40-3	12-3	126-0	1 L.P. turbine, 1 recip'g engine.	1 H.P. turbine.....	1 L.P. turb., 1 recip'g engine ..	Parsons.....	710	260
218	175	1,036	...	300-0	73-6	40-3	12-3	126-0	1 L.P. turbine, 1 recip'g engine.	1 H.P. turbine.....	1 L.P. turb., 1 recip'g engine...	Parsons.....	710	260
218	175	1,036	...	300-0	73-6	40-3	12-3	126-0	1 L.P. turbine, 1 recip'g engine.	1 H.P. turbine.....	1 L.P. turb., 1 recip'g engine...	Parsons.....	710	260
213	170	1,046	16,500	300-0	73-6	40-3	12-3	126-0	1 turbine, 1 recip'g engine.....	.....	1 turbine, 1 recip'g engine	Cramp	620	260
213	170	1,046	16,500	300-0	73-6	40-3	12-3	126-0	1 turbine, 1 recip'g engine.....	.....	1 turbine, 1 recip'g engine	Cramp	620	260
213	170	1,046	16,500	300-0	73-6	40-3	12-3	126.0	1 turbine, 1 recip'g engine.....	.....	1 turbine, 1 recip'g engine	Cramp	620	260
218	175	1.036	...	300-0	73-6	40-3	12-3	126-0	1 L.P. turbine, 1 recip'g engine	1 H.P. turbine.....	1 L.P. turb., 1 recip'g engine..	Parsons.....	710	260
213	170	1,046	6,500	300-0	73-6	40-3	12-3	126.0	1 turbine, 1 recip'g engine.	.....	1 turbine, 1 recip'g engine	Cramp	620	260

akwater course, deduct $4,000.

† If tried on Delaware Breakwater course, deduct $4,900.

LS OPENED AUGUST 7, 1911.

DEPARTMENT DESIGN.
MENT ALTERNATE PROPOSAL.

rbines.	Revolutions at designed power.	Steam pressure.		Reciprocating engines.			Operating through speed-reduction gear.		Boilers.			Air pressure in fire-room, maximum.	Cooling surface, main condensers, total.	Bureau S. E. penalty weights.		
		At boiler.	At engine.	Revolutions.	Cylinder diams.	Stroke.	Maxi. S.H.P.	Revolutions.	Type.	Number.	Heating surface, total.			Boilers.	Machin. and appurt.	Total S. E. weights.
		p.g.	*p.g.*	*ins.*	*ins.*						*sq. ft.*	*ins.*	*sq. ft.*	*tons.*	*tons.*	*tons.*
.....	710	260	240	340 about	...	...	...	...	Normand	4	21,500	5	10,750	...	...	326
.....	700	260	240	345	12½-17½	14	None	none	Thornycroft express	4	21,500‡	6½	12,000	99.6	226.4	326
.....	700	260	240	None	None	none	8,000	550	Thornycroft express	4	21,500‡	6½	12,000	99.6	226.4	326
.....	700	260	240	None	None	none	None	none	Thornycroft express	4	21,500‡	6½	12,000	99.6	226.4	326
.....	550	265	240	350	12½-26½	13½	...	...	Thornycroft express	4	21,500‡	6½	12,000	123.5	263.95	387.45
.....	547	265	250	...	...	...	...	...	Yarrow water tube..	4	21,500	5	10,200	...	...	321.22
.....	547	265	250	253 at 15½ k.	12½-26½	13½	...	...	Yarrow water tube..	4	21,500	5	10,200	...	...	326
s, cruis-ghouse.																
.....	710	260	240	345	12½-17½	14	...	...	Thornycroft.....	4	21,500	5	11,600	130	199	329
.....	710	260	240	...	...	...	1,590	345	Thornycroft.....	4	21,500	5	11,600	130	204	334
.....	550	265	250	260	12½-26½	13½	...	...	Thornycroft.....	4	21,500	5	11,600	130	207	337
.....	...	265	250	...	...	...	...	...	Thornycroft.....	4	21,500	5	11,600	130	199	329
.....	710	260	240	380	11-21	12	...	...	Express.....	4	21,500	6	11,600	102.5	239.5	342
.....	710	260	240	380	11-21	12	...	...	Express.....	4	21,500	6	11,600	102.5	239.5	342
.....	710	260	240	380	11-21	12	...	...	Express.....	4	21,500	6	11,600	102.5	239.5	342
.....	620	260	240	300	12-23	12	...	...	Express.....	4	21,500	6	11,600	102.5	249.5	352
.....	620	260	240	300	12-23	12	...	...	Express.....	4	21,500	6	11,600	102.5	249.5	352
.....	620	260	240	300	12-23	12	...	...	Express.....	4	21,500	6	11,600	102.5	249.5	352
.....	710	260	240	380	11-21	12	...	...	Express	4	21,500	6	11,600	102.5	239.5	342
.....	620	260	240	300	12-23	12	...	...	Express	4	21,500	6	11,600	102.5	249.5	352

‡ Or more.

DENMARK.

The torpedo boat *Soridderen*, built for the Royal Danish Government by Messrs. Yarrow & Co., Glasgow, ran her official full-speed trial on June 24th on the Skelmorlie deep-water measured mile at the mouth of the Clyde. The contract speed of 27 knots was easily attained during a three-hours' continuous trial, with the vessel carrying a load of 50 tons.

The *Soridderen* is 181 feet 9 inches long by 18 feet beam, and is propelled by twin screws driven by turbines of the Brown-Curtis type, constructed by Messrs. Yarrow. Steam is supplied by two Yarrow water-tube boilers of the latest type.

GERMANY.

The scout *Augsburg*, built at Kiel, recently completed her trials. Speed, 27¼ knots, with 31,000 S.H.P.

She is the third of the *Mainz* type, and has Parsons turbines, the *Mainz* having Curtis, and the *Kolberg*, Melmes and Pfenninger. Her boilers, fifteen in number, are of the Schulz-Thornycroft type. Bunker capacity, 900 tons.

One set of Diesel engines, of 6,000 H.P., forming half of the 12,000 H.P. installation for a German war vessel; have been under test at Nuremberg. These engines have three cylinders, 2-cycle, double acting; revolutions, 145 to 150.—"Rivista Maritima."

OBITUARY.

EDWARD D. ROBIE.

"To the past go more dead faces,
Every year;
As the loved leave vacant places,
Every year;
Everywhere the sad eyes greet us,
In the evening's dusk they meet us,
And to come to them entreat us,
Every year."

Chief Engineer Edward D. Robie was born in Burlington, Vermont, in September, 1831, and received an academic education in that city, besides learning the trade of machinist.

At the age of 21 he entered the Navy as a Third Assistant Engineer and was ordered to the steam sloop *Mississippi*, the flag ship of Commodore Perry's expedition to Japan, and served on board that vessel for three years. He was a versatile young man, ingenious and resourceful, and of his services Commodore Perry made great use during that period.

The Perry expedition conveyed many presents to the Mikado, and skilled artisans and artists were carried in the ship for the purpose of demonstrating and teaching the Japanese the arts and discoveries of Americans and to enter into commercial relations with them. Rear Admiral Upshur and Chief Engineer Robie are the last survivors of that famous expedition.

Among the experts on board was an artist who had studied the art of Daguerre—for it was before the days of photography—but who, for some reason, failed to produce a picture. Commodore Perry assigned Mr. Robie to the task, and his effort was crowned with success. Mr Robie erected and

operated the first telegraph line in Japan; he afforded material assistance in building the first railroad and in operating the first steam locomotive in that Hermit Kingdom.

Commodore Perry has been called the father of the steam Navy; he had one or more of his engineers with him whenever he visited a port, not that he had any particular liking for them, but because they were useful to him. They had brought into the service arts and methods hitherto unknown, and were resourceful in many ways.

Mr. Robie was promoted to Second Assistant Engineer in 1855, and served on board the *Michigan* for a year, when he was transferred to the *Susquehanna,* which vessel was the convoy of the *Niagara* when she laid the first Atlantic submarine cable in 1856–7. The *Susquehanna* then went to Nicaragua to capture the famous Walker filibuster expedition, where an epidemic of yellow fever so depleted the crew that the vessel was obliged to return to New York.

Mr. Robie was promoted to First Assistant Engineer in 1858, and went, in the *Niagara,* to the West Coast of Africa to return captured negro slaves. The slave trade was at that time quite active, and the United States, England and France kept war ships on that coast to break up the barbarous traffic, but no other nations gave any assistance.

In 1859 Mr. Robie joined the *Lancaster* at Philadelphia, which went to the Pacific, via Cape Horn, and, on reaching Panama, was transferred to the *Saranac* as Senior Engineer, that vessel then going in search of the sailing sloop of war *Levant,* which was missing.

Mr. Robie was promoted to Chief Engineer in 1861 and was ordered to the *Mohican,* a screw ship. The Civil War being then in progress, he served in the *Mohican* in the battles at Port Royal, Brunswick, Ga., and Fernandina, and returned to New York in 1862, where he was detailed to superintend the building of the iron-clad monitor *Dictator,* her hull armor and machinery complete. At that time the experience of the naval constructors was limited to wooden shipbuilding only.

In 1864 Mr. Robie took charge of the operations of the

steamship *Ericsson*, raising obstructions in the harbor of Charleston, after which he was on duty as member of several Boards of Inspection and on court martial duty. In 1865 he was on the Engineer Board of Examiners. Here he endeared himself to those going up for examination for promotion by his very clear and comprehensive questions, and by his uniform kindness to candidates.

In 1866 Mr. Robie went, in the *Ossipee*, to the Pacific, via the Straits of Magellan, and on arrival at Panama was promoted to Fleet Engineer, on board the flag ship *Pensacola*. In 1869 Mr. Robie served on the famous Board of which Rear Admiral L. M. Goldsborough was president, and made a minority report against the general condemnation of ships which the Board recommended. After 42 years of experience and improvement in the art of ship building and ship designing, his minority report stands out in favorable contrast with that of the majority.

Mr. Robie was at the Boston Navy Yard in 1870–1, and, when ordered to the *Wabash*, he improvised a steam-steering engine for that frigate, which was the first successful steam steerer used in the Navy. The cruise of the *Wabash* was in the Mediterranean, with Commodore Alden as Flag Officer and with General W. T. Sherman and his staff as guests on board. Subsequently Mr. Robie was Fleet Engineer on the North Atlantic and Gulf Squadrons, and then, from 1874 to 1877, was Chief Enginner of the Norfolk Navy Yard. He did special inspection duty at Pittsburg and Cold Spring foundry, N. Y., and was again Fleet Engineer in the Pacific on the Flag Ship *Pensacola*; then Chief Engineer of the Boston Navy Yard; then of the New York Yard and of the Norfolk Yard. His last duty was at the Navy Department in Washington, where he remained until retired for age in 1893. During the war with Spain Mr. Robie was employed selecting vessels for the auxiliary naval force at Charleston, Savannah, Baltimore and New York.

His sea service aggregated 16 years and 11 months and his shore duty 21 years and 9 months. He was retired with the

rank of Commodore, but by an act of Congress was promoted to the rank of Rear Admiral.

Personally Rear Admiral Robie was a most attractive, agreeable man, always popular on board ship, and never more happy than when doing a kindness to someone. He was a member of Washington Commandery of the Loyal Legion, and of Otsemingo Lodge of Freemasons, at Binghampton, N. Y.

He died peacefully as he had lived at his home in Washington on the 7th of June, 1911. He had reached the age of 80 years, and had simply worn out. The immediate cause of his death was uremic poisoning, following pneumonia.

He leaves a widow, one son and two daughters, one of whom is the wife of Surgeon A. G. Grunwell, of the Navy.

Rear Admiral Robie was buried at Arlington National Cemetery on the 9th of June.

G. W. B.

RICHARD INCH.

Rear Admiral Richard Inch, who died at Washington on the 21st of April, 1911, was born in Washington, in 1843, and was educated in the public schools of that city. He learned the trade of machinist in the Navy Yard, and became a draughtsman there. He entered the Navy as a Third Assistant Engineer 8th September, 1863, in a class of fifteen, and was by competition number three of his date. For that date there were 95 competitors, of whom but 15 were accepted.

Mr. Inch was assigned to the *Lancaster*, in the Pacific, where he remained until 1866. He was promoted to the grade of Second Assistant Engineer in 1864; to Passed Assistant Engineer in 1874, and to Chief Engineer in 1892; was transferred to the line of the Navy by the Personnel Act, approved 3d of March, 1899, as a Commander; promoted to the rank of Captain in 1902, and retired as a Rear Admiral in 1905.

He served on board the *Lancaster*, the *Nyack*, *Gettysburg*, *Wyoming*, *Passaic*, *Pinta*, *Yantic*, again on the *Lancaster*, the *Marion*, *Charleston*, and on board the *Boston*. He was

Chief Engineer of the last named vessel at the battle of Manila Bay. His sea service aggregated 21 years and 4 months. He served on shore in the Navy Yard at Washington two terms, was inspector of coal under the Bureau of Equipment, served at the Mare Island Yard and as Inspector of Machinery at Newport News, Va. He was advanced three numbers, by act of Congress, for services in the battle of Manila Bay.

Rear Admiral Inch was identified with the Navy from childhood. His father was an employee in the Navy Yard; his elder and only brother was a chief engineer in the Navy, and his brother-in-law a master mechanic in the Navy Yard, who began his career thus about 1840.

Rear Admiral Inch was a member of the Loyal Legion and of the American Society of Naval Engineers. He was a man of strictly moral habits, abstemious in all things, and had a host of friends and few, if any, enemies.

G. W. B.

JOSEPH TRILLEY.

Rear Admiral Joseph Trilley was born in Ireland in 1837, and came to Baltimore when a child. He was educated in the public schools of that city and learned the trade of machinist at Pool and Hunt's in that city. He entered the Navy as a Third Assistant Engineer in 1860. He served on board the *Pawnee* and was in the fights at Fort Sumter, Norfolk, Aquia Creek and several minor engagements in the Potomac River; he was at the capture of Hatteras Inlet, Hilton Head, Fernandina, and in the engagement at Stone Inlet. He was on board the *Monongahela* during the engagements at Port Hudson, Donaldsonville and at Mobile Bay. He served on board the *Algonquin* and the *Idaho* during their trial trips, and later on board the *Shawmut, Ossipee, Vandalia, Wabash, Trenton, Monterey, Olympia* and *Philadelphia,* his sea service aggregating 17 years. Rear Admiral Trilley was transferred to the line of the Navy by virtue of the act approved March

3d, 1899, as a Captain, and was retired in 1905 as a Rear Admiral. His services on shore were at the Navy Yard, Boston; the Rendezvous at Boston; experimental duty at Boston; also at Portsmouth and Mare Island Navy Yards as Chief Engineer. He died at his home in San Francisco on the 6th of March, 1911, leaving a widow, but no children.

Rear Admiral Trilley was a man of strictly moral habits, industrious, painstaking and faithful to his trust. He was always popular with his shipmates.

G. W. B.

BOOKS RECEIVED.

MARINE ENGINE DESIGN. Including the Design of Turning and Reversing Engines. By EDWARD M. BRAGG, S. B., Assistant Professor of Marine Engineering, University of Michigan. 172 pages. D. VAN NOSTRAND Co., New York. $2.00 net.

This work contains all of the essential information necessary for the design of reciprocating marine engines, reversing engines and turning engines. The historical features of most works on the marine engine, with descriptions of various types that are more or less obsolete, are omitted. The book is written for engineers who know the marine steam engine and desire, separated from the chaff to be found in other books, a reference book that will give them the proper sizes of cylinders and the proportions of the various parts. The book meets this want very successfully. The formulas for design are given, with tables showing the best practice, and there is sufficient theory to meet the wants of the careful designer who aspires to better work than is given by thumb rule.

MARINE GAS ENGINES. THEIR CONSTRUCTION AND MANAGEMENT. By CARL H. CLARK, S. B. 115 pages, 102 illustrations. D. VAN NOSTRAND Co., New York. $1.50 net.

This little book treats of marine motors of the internal-combustion type using volatile fuels. Engines of different types are described, as well as vaporizers and carburetors, ignition devices, wiring and lubrication. There is a chapter each on the subjects of multiple-cylinder engines, reversing mechanisms, propellers, installation, operation and care of engines, power of engines and selecting an engine.

The text is clear and comprehensive and there is no space

wasted. The cuts are excellent and add much to the value of the book.

This book is well adapted to the needs of officers and others who desire to become better acquainted with engines for motor boats.

COLD STORAGE, HEATING AND VENTILATION ON BOARD SHIP. By SYDNEY F. WALKER, R. N. 269 pages. D. VAN NOSTRAND CO., New York. $2.00 net.

The cold-storage problem for meats and other perishable stores is here treated; also, the proper temperatures to be carried, the design of cooling compartments, with method of covering the walls, the circulation of refrigerating medium, etc. The different types of refrigerating machines that are employed for marine use are discussed and their efficiencies compared. Considerable space is given to the faults that are encountered in the operation of refrigerating machinery.

The part relating to heating and ventilation discusses the various methods employed for heating vessels, with illustrations of the various kinds of heating apparatus. The thermo-tank system is discussed at length, leading up to the subject of ventilation. The methods employed in many large vessels are given.

A SHORT HISTORY OF THE UNITED STATES NAVY. By OFFICERS OF THE ENGLISH DEPARTMENT, U. S. NAVAL ACADEMY.—This book is a compilation from various sources, containing 479 pages, with numerous illustrations. It has been written with certain definite aims, the principal of which are the following: To present a record of naval events exactly as they occurred, without prejudice of any kind; to regard such events from the professional rather than the picturesque point of view: for example, to emphasize the maneuvering and gunnery in an action rather than the smoke and blood; to suppress the trivial and bring out the important, not neglecting

the services of officers in time of peace, and to give, whenever possible, the original sources of information, such as letters from Secretaries of the Navy, and official reports.

The history of the Navy is here presented in condensed form, but the principal events are all recorded and the narrative, which is written in attractive style, can be relied upon as presenting an accurate account of the Navy from its beginning to the present time.

The book will be found of much value to all who are interested in the Navy. It is published by the J. B. LIPPINCOTT COMPANY, Philadelphia. Price, $3.00.

ASSOCIATION NOTES.

Lieutenant Commander John Halligan, Jr., U. S. N., has been elected a member of the Council to fill the vacancy caused by the resignation of Commander John K. Robison, U. S. N., the change taking effect July 1, 1911.

The following Members and Associates have been received into the Society since the publication of the list of additional members in the May number of the JOURNAL.

MEMBERS.

Hoover, J. H., Ensign, U. S. Navy.
Smeallie, J. M., Lieutenant, U. S. Navy.

ASSOCIATES.

Andrew, S. Alfredo, Engineer Lieutenant, Argentine Navy, Hotel Greenleaf, Quincy, Mass.

Carcagno, Juan A., Engineer Lieutenant, Argentine Navy, 1572 Hancock Street, Quincy, Mass.

Frahm, H., Director, Blohm & Voss, Hamburg, Germany.

Lange, W. H. A., Bureau of Steam Engineering, Navy Department, Washington, D. C.

Meehan, J. H. B., Chief Engineer, N. A. S., U. S. S. *Vulcan*, care of Postmaster, New York City.

Moise, B. C., Assistant Secretary-Treasurer and Auditor, National Tube Company, 5320 Ellsworth Avenue, Pittsburgh, Pa.

Spyer, Arthur, Manager Marine Department, Babcock & Wilcox, Ltd., London, E. C., England.

The membership of the Society is steadily increasing, but there are many officers in the service who are not members and who would be glad to join if the desirability of doing so were brought to their attention. Members are urged to co-operate in bringing into the Society all officers who are interested in the development of engineering in the Navy.